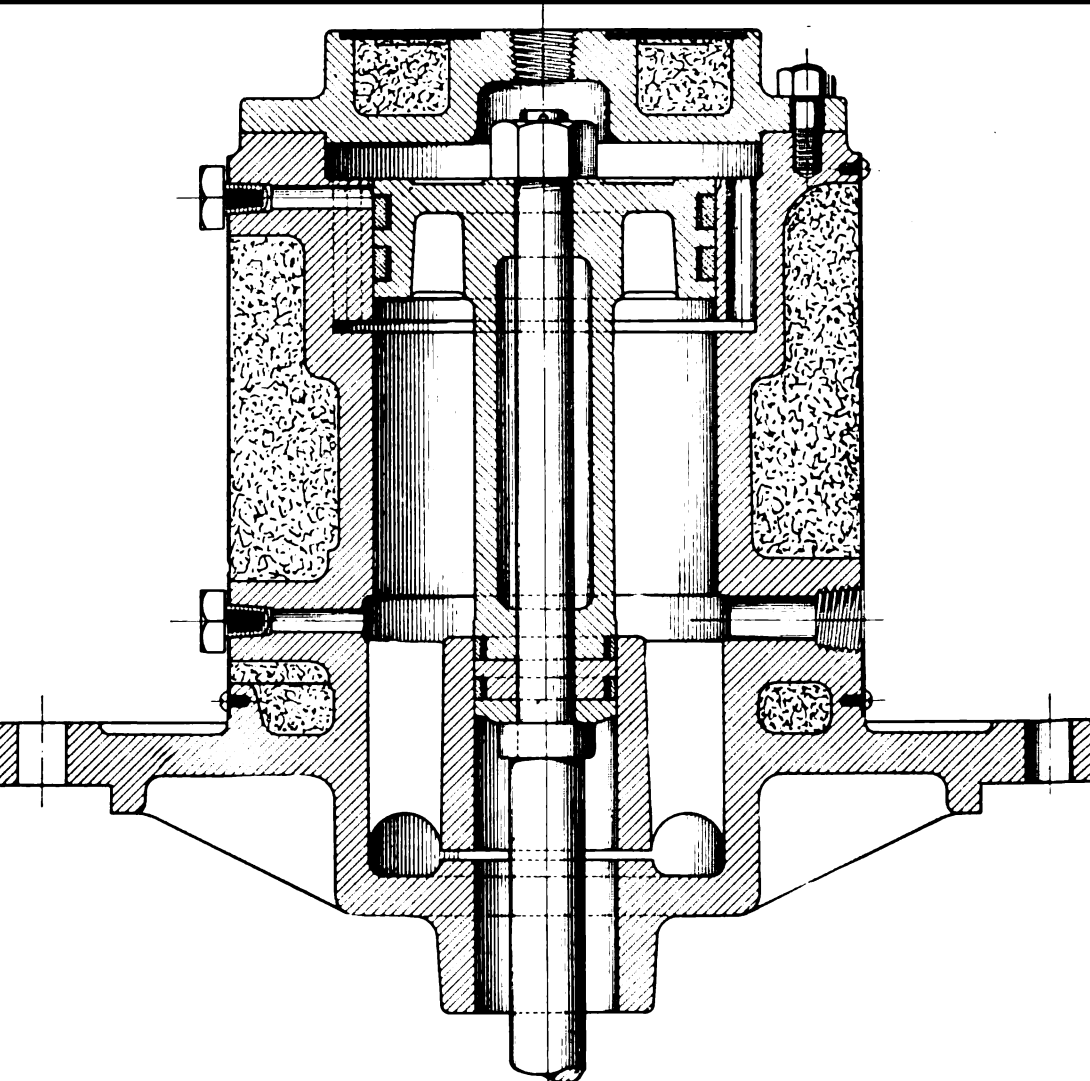

Journal of the American Society of Naval Engineers, inc

American Society of Naval Engineers

JOURNAL

OF THE

American Society of Naval Engineers

10459

PUBLISHED QUARTERLY

UNDER THE SUPERVISION OF THE COUNCIL.

VOLUME XIX.

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WASHINGTON, D. C.
R. BERESFORD, PRINTER, 618 F STREET, N. W.
1907.

INDEX TO VOLUME XIX, 1907.

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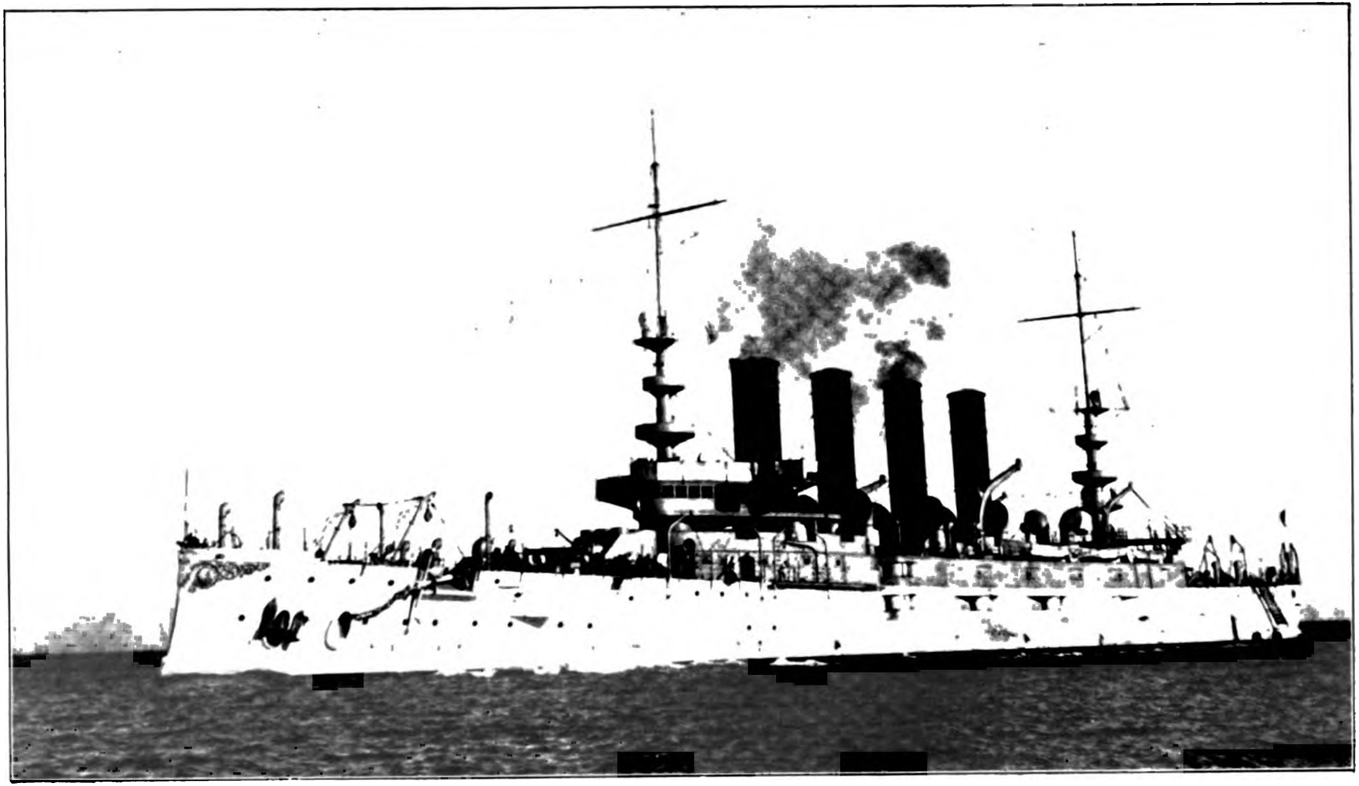
Illustrations:



U. S. S. "CALIFORNIA."

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XIX. FEBRUARY, 1907. No. 1.

U. S. S. *CALIFORNIA*.

DESCRIPTION AND OFFICIAL TRIALS.

BY G. W. DANFORTH, LIEUTENANT, U. S. NAVY, MEMBER.

The *California*, armored cruiser No. 6, belongs to a class of six vessels authorized by Congress in June, 1900. The other vessels of this class are the *Colorado*, *Pennsylvania*, *Maryland*, *West Virginia* and *South Dakota*. The contract for the *California* was signed by her builders, the Union Iron Works, San Francisco, on January 10, 1901, requiring her completion in thirty-six months, at a price of $3,800,000 exclusive of armor, ordnance, boats, portable furniture and lesser outfit usually furnished by the Government. The time for completion was extended in three stages to June 30, 1906, due to a prolonged labor strike, delays in reorganizing the labor forces after the strike, changes in plans and specifications, and other causes of less import; and a further extension is asked for in consequence of the earthquake and fire

which prostrated San Francisco in April, 1906. The most important change in the machinery was that necessary for outboard instead of inboard-turning propellers, this having been made before the engines were far advanced. The original contract provided for sheathing and coppering the hull, but this requirement was canceled. The hull was launched April 28, 1904, and the machinery was installed after launching. The dock trials of the engines took place from January 30 to February 8, 1906, and were entirely satisfactory.

The contract requires that an average speed of 22 knots per hour shall be maintained for four consecutive hours, with heavy penalties for failure to make this speed, but no premiums are allowed, as in former practice, for any excess of speed over 22 knots. The successful making of this speed insures preliminary acceptance only. Final acceptance is reserved until the vessel is tested in actual service, but this test must take place within six months after preliminary acceptance.

HULL DATA.

The principal hull data are as follows:

Length between perpendiculars, feet	502
on load water-line, feet	502
over all, feet and inches	503-11
Beam, extreme, feet and inches	69-10¼
at load water-line, feet and inches	69- 6¼
Molded depth, main deck, feet	41
bridge deck, feet and inches	48- 6
Normal mean draught, feet and inches	24- 1
Displacement at normal draught, tons	13,680
full load draught (full bunkers), tons	15,138
per inch of draught at load water-line, tons	57.8
Wetted surface, square feet	43,350
Coefficient of fineness, block, per cent	57
midship section, per cent	95
of load water-line, per cent	69
Cylindrical coefficient, longitudinal, per cent	60
Number of frames	124
water-tight compartments	279

The bridge and main decks are laid with teak, the gun deck with linoleum, except in certain spaces like galley and toilet rooms, and the berth deck has no linoleum in the passages nor wash rooms.

The thickness and disposition of armor is the same as on vessels of this class, accounts of which are published in previous issues of this JOURNAL.

BATTERY.

No. of guns.	*Caliber or kind.*	*Gun mark.*	*Remarks.*
4	8-inch	V	Forty calibers long.
14	6-inch	VIII	Fifty calibers long.
18	3-inch	III	
12	3-pdr.	IV	
2	1-pdr.	V	
2	3-inch field gun	I Mod. I	
2	Gatling		
2	Colt		

There are fitted two stationary torpedo tubes under the protective deck, forward, to discharge below the water line.

MACHINERY.

The vessel has two main engines, each four-cylinder, triple-expansion, vertical, inverted, placed abreast in separate watertight compartments. These engines were designed for a horsepower of 23,000, when making about 120 revolutions per minute and with a steam pressure of 250 pounds at the high-pressure cylinder. The order of arrangement of cylinders, beginning forward, is F.L.P., H.P., I.P., A.L.P., and the cranks, at 90-degree intervals, are in the following sequence, H.P., I.P., F.L.P., A.L.P. The cylinders are supported by forged columns of mild steel, braced diagonally with 'thwartship and fore-and-aft bracing of the same material. Each engine crank shaft is forged in two pieces, each piece with cranks at 180 degrees, and the turning engine is attached by worm and tooth gearing to the flanges joining these shafts. The reversing shaft is located on the inboard side of each en-

gine, near the top of the framing. The Stephenson double-bar link valve gear is used.

Steam is supplied by sixteen water-tube boilers in eight watertight compartments, a center-line bulkhead separating the port and starboard pairs of boilers connected to a common smoke funnel. The eight boilers on each side of the ship have their steam spaces connected to a line of 8-inch piping which extends fore and aft over the boilers, throughout the fireroom spaces; the port and the starboard lines connecting in the forward firerooms, from whence branches carry steam to the dynamos and other auxiliaries located forward, and also connecting in the engine rooms, where branches supply adjacent auxiliaries. The main steam pipe, extending from the engines forward only over six boilers on each side, is not connected directly to the boilers, but is fed by the 8-inch line, which practically parallels it, through connections therewith controlled by stop valves. These connections are at three points on each side of the fireroom spaces, at one point on each side of the compartment between fire and engine rooms, and at the main separators. All this piping is of steel, seamless-drawn, with welded flanges.

MAIN ENGINE DATA.

	H.P. *inches.*	I.P. *inches.*	L.P. *inches.*
Diameter of main cylinders	38½	63½	74
Stroke, all cylinders	48	48	48
Valves, one piston valve for each H.P. cylinder, diameter	24½	...	...
two piston valves for each I.P. cylinder, diameter	27	...	...
one double-ported flat valve for each L.P. cylinder, dimensions	...	...	62½×80½
travel	10	10	10
Balance valve pistons, diameter	7	6	16
Valve stems, diameter	3½	3½	4
through valves	2½	2½	2 7/16
at balance pistons	1¾	1¾	1¾
Main steam pipe, diameter at throttle	13	...	...
Piston rods, diameter	8½	8½	8½
of axial holes	4½	4½	4½
length	118	118	118

	H.P. *inches.*	I.P. *inches.*	L.P. *inches.*
Cylinder walls, thickness	2¼	2	1¾
liners, thickness	1¾	1¾	1⅛
diameter of studs in head	1⅝	1⅝	1⅝
pitch of studs in head	4	5¼	6
Port areas, minimum	195	438	495

Connecting rods, length center to center, feet	8
diameter, upper end, inches	8¼
lower end, inches	9¼
axial hole, inches	5
crosshead bolts (4), inches, outside,	3⅛
axial hole, inches	1⅝
crankpin bolts (2), inches, outside	4¾
axial hole, inches	2
Main bearings, length, five of 22; one (between I.P. and A.L.P.) inches	35
Main bearing bolts (four), diameter, inches	3
Crosshead, surface, ahead, square feet	5.625
backing, square feet	2.64
pins, diameter, inches	11
length, each, inches	11
Speed of pistons, in feet per minute, at 120 revolutions	960
Weight of pistons, complete, each, H.P., pounds	1,933
I.P., pounds	3,009
L.P., pounds	3,489
piston rods, each, pounds	2,411
crossheads, each, pounds	2,338
connecting rods, each, pounds	7,165
Crank shaft, diameter, inches	19⅛
of axial holes, inches	10
coupling disks, inches	32
thickness coupling disks, inches	4⅛
coupling bolts (10 in one flange), diam., inches	3⅛
pin, diameter, inches	20
length, inches	23
diameter axial hole, inches	12
webs, width, inches	21
thickness, inches	12
Thrust shaft, diameter at bottom of collars, inches	17¾
axial hole, inches	10⅛
length, feet	17⅛
collars, number on each shaft	13
diameter, inches	27
thickness, inches	2¼
distance between, inches	5
total thrust surface each shaft, sq. feet	29.35
actual bearing surface of horse shoes, each shaft, square feet	14.26

Thrust-bearing side rods (two to each bearing), diameter, each, inches	4⅛
Line shaft, diameter, inches	17⅛
axial hole, diameter, inches	10⅛
coupling discs, diameter, inches	33⅛
coupling bolts (ten in each flange), diameter, inches	3⅛
length (one length each engine), inches	241
number of bearings to each length	1
length of bearing, inches	27
weight of shaft, pounds	11,900
Stern-tube shaft, diameter, without casing, inches	18⅛
axial hole (at after end, 7 inches), inches	10⅛
length (one to each engine), inches	490⅛
weight, pounds	25,600
steel coupling sleeve, diameter, inches	26
cotters (one in each end of sleeve) thickness, inches	3⅛
width at ends, inches	9 and 10⅛
Propeller shaft, diameter, inches	18⅛
axial hole, diameter, forward end, inches	7
along shaft, inches	10⅛
after end, inches	9
length (one to each engine), inches	570⅛
Reversing engine, diameter, steam cylinder, inches	16
controlling cylinder, inches	9
stroke, inches	22
Turning engine, two steam cylinders, diameter, each, inches	7
stroke, inches	6

Clearances, main engines, expressed in per cent. of stroke:	*Top.*	*Bottom.*
Starboard high-pressure	24.91	22.57
Port high-pressure	25.34	23.16
Starboard intermediate-pressure	22.25	20.24
Port intermediate-pressure	22.33	20.49
Starboard forward low-pressure	13.19	12.68
Port forward low-pressure	13.65	12.29
Starboard after low-pressure	13.15	12.53
Port after low-pressure	13.08	12.52

VALVES AND VALVE SETTINGS.

	H.P. cylinder.		I.P. cylinder.		F.L.P. and A.L.P. cylinders, each.	
Number of valves, one engine	1		2		1	
Type	Piston		Piston		Double-p't'd slide	
Diameter of valves, inches	24½		27		...	
Travel of valves, inches	10		10		10	
Side of valve on which steam is taken	Inside		Outside		Outside	
Angular advance, degrees and minutes	25-10		38-45		45-30	
Width of port, and length of slide valve, inches	4½		4½		4×67½, double	
	Top	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom.*
Steam opening, linear inches	3⅛	3 27/32	2 11/16	2⅞	2 of 2 5/16	2 of 2 7/16
area of sq. in.	237.86	260.96	407.80	435.20	304.84	335.34
Exhaust opening, inches	3 29/32	3 11/16	full pt.	4	full port	3¾
area sq. in.	265.04	250.09	682.38	606.56	552	517.5
Steam lap, inches	1⅛	1 5/32	2 5/16	2⅛	2¾	2 9/16
Exhaust lap, inches	1 3/32	1 5/16	⅛	1	⅛	1⅛
Steam lead, linear inches	⅝	21/32	13/16	29/32	13/16	13/16
Cut-off, in decimal of stroke, maximum, per cent	89.06	86.18	76.3	65.1	66.5	55
Exhaust release, inches from beginning of stroke	47¼	47¼	37¼	44⅛	42⅛	40⅝
Compression, inches from end of stroke	6⅝	4⅛	9⅛	6⅝	8¼	6¾
Steam admission, inches before end of stroke	9/16	⅝	13/16	5/16	11/16	⅛

MAIN ENGINE AUXILIARIES.

There is a condenser for each main engine, located outboard of the engine, cylindrical, with steel shell 7 feet 5 inches in diameter, and composition heads. There are 6,280 seamless-drawn Muntz-metal tubes in each condenser, 14 feet 2 inches long, ⅝ inch outside diameter, and No. 18 B.W.G. in thickness, tinned inside and outside. These give each condenser a cooling surface of 14,391 square feet. Each condenser is supplied with cooling water by a centrifugal pump, driven by a vertical cross-compound engine. The pump has double inlets, a vane 48 inches in diameter, and the suction and discharge openings are 21 inches in diameter, with a suction connection to the main drain 15 inches in diameter. The engine has a stroke of 12 inches, the high-pressure cylinder is 9 inches, and the low, 12 inches in diameter. At a speed of 180 revolutions per minute on the standardization trial each circulating engine developed 65 I.H.P. These pumps were built by the Union

Iron Works, all other principal pumps of the vessel having been built by other builders on the Pacific coast.

To each main condenser is attached an independent air pump of the usual design supplied to vessels to the U. S. Navy. The steam cylinders are 14 inches, and the air cylinders 35 inches in diameter, with a common stroke of 18 inches. Each of these pumps developed 13 I.H.P. at 20 revolutions per minute on the vessel's standardization run.

BOILERS.

Steam for all uses of the vessel is supplied by sixteen Babcock & Wilcox water-tube marine boilers placed in eight watertight compartments. These boilers are placed with fronts facing athwartship firerooms, the two boilers of each compartment having a common firing space 12 feet wide. Each fire room is connected directly with adjacent bunker, outboard by a watertight door, and each adjacent pair of port and starboard firerooms is connected by a similar door, through the middle-line bulkhead, which extends throughout the engine and boiler spaces. Fireroom passageways are convenient, and spaces around and above the boilers roomy.

All pressure parts of these boilers are of open-hearth steel plate and cold-drawn seamless steel tubes. All tubes are two or four inches in diameter, and tube ends are expanded into headers. The longitudinal seam of each steam drum is butted, and strapped inside and outside. There are four rows of rivets through both straps and two rows additional through the shell and inside strap.

The boiler casings are airtight, and consist of fire brick, $\frac{1}{4}$-inch asbestos millboard, and 2 inches of magnesia block backed with galvanized steel plates about $\frac{1}{16}$-inch in thicknes braced with angle irons and bolted to the boiler foundations. Openings are provided in casings for cleaning. The fire fronts and fire and ash doors are of wrought-steel plate. All pressure parts were tested to an internal hydrostatic pressure of 500 pounds per square inch at the works of the builders. The principal boiler data are as follows:

Working pressure, per gauge, pounds per square inch	265
Heating surface, all boilers, square feet	70,928
Grate surface, all boilers, square feet	1,591.36
Length of grates, feet	7
Ratio of heating to grate surface	44.57
Diameter of opening of safety valves (three each boiler), inches,	4
Thickness of 2-inch generating tubes, B.W.G	8
4-inch generating tubes, B.W.G	6
Outside diameter of drum, inches	42
Thickness of drum plate, inches	$\frac{21}{32}$
Tensile strength of drum plate, pounds per square inch	60,000
Thickness of convex drum heads, inch	$\frac{3}{4}$
Number of boilers per smoke stack	4
Height of smoke stack above grate bars, feet	93
Area through smoke stack, square feet	58.99
Ratio grate surface to stack area	6.7
Weight of boiler dry, ready for service, pounds	95,891
water, one boiler, at steaming level and temperature (410 degrees), pounds	15,214

BOILER FEED SYSTEM.

For use in feeding the boilers there are four main feed pumps, exclusively for this use, and eight auxiliary feed pumps connected also for fire and bilge service and for pumping out boilers.

The main feed pumps are simple, double-acting, vertical, with central-packed plungers. Dimensions are, steam cylinders, 14 inches diameter; water cylinders, 10 inches diameter; stroke, 18 inches. Two of these pumps are placed on the forward bulkhead of each engine room, all draw water only from the pipe connecting the two main feed tanks, and each pair delivers water primarily to the boilers on one side of the ship, but a cross connection of the feed mains in the forward fire-rooms enables any of these pumps to supply any boiler.

The auxiliary feed pumps, one in each boiler compartment, are of the same type as the main feed pumps, with 12-inch steam cylinders, 8-inch water cylinders and 12-inch stroke. Each auxiliary pump is connected to feed any boiler on its own side of the ship.

A feed-water tank is built outboard in each engine room, and these tanks, each having a capacity of 6,500 gallons,

receive the discharge from all main and auxiliary air pumps, traps and drains of steam lines. The tanks are arranged for using filtering material, and each has a pipe to conduct vapor to the atmosphere. To each tank is attached a hot-well pump for handling feed water. Each of these pumps has suction connection on its own side of the ship with the main air-pump suction pipe, the feed-water tank, the reserve feed-water tanks and the ship's side above the water line, for receiving fresh water from outside sources. The dimensions of these pumps are 12 × 16 × 18 inches.

The reserve feed-water tanks occupy three double-bottom compartments on each side of the keel, under the after fire-rooms, and their capacity is about 150 tons of fresh water.

The main feed pumps in each engine room discharge through a feed-water heater which heats with steam from the auxiliary exhaust. This discharge passes also through an oil separator in each engine room. There are by-passes to these heaters and separators.

PROPELLERS.

The two propellers, outboard turning, each with three adjustable blades, are made of manganese-bronze, from Union Iron Works design. Each is a true screw, and the blades were carefully surfaced, balanced and measured for pitch by an especially-constructed machine, before leaving the shops. Hubs and blades are tinned. Each hub is fitted to the tapered end of its shaft, and is held in place by a steel key with rounded ends, fastened along the tapered part of the shaft, also by a nut on the end of the shaft, threaded opposite to the direction of ahead rotation of the propeller. This nut is itself secured against unscrewing by a special locking device. The principal data of the propellers are as follows:

Diameter of propeller, feet	18
hub, feet	5
Disc area, square feet	254.4
Helicoidal area, square feet	95
Ratio mean pitch to diameter	1.1944

Pitch is adjustable between 20 and 23 feet.
Distance of hub center above lowest point of keel, feet and inches, 10-00$\frac{3}{8}$
Weight, complete, starboard propeller, pounds........................ 34,499
port propeller, pounds............................... 34,820

OTHER STEAM AUXILIARIES AND EQUIPMENT.

In addition to the auxiliaries previously named, the ship is provided with others, as below noted:

The steering engine is of the usual type fitted on U. S. naval vessels, and was built by the Union Iron Works.

The boilers are fitted with the Howden system of forced draft. Air is supplied by a blower over each boiler, with a fan 69 inches in diameter, driven by a compound engine of 6-inch stroke and with cylinders 5 and 8 inches in diameter, respectively.

In each engine room is located an auxiliary condenser, with air and circulating pumps, receiving exhaust steam from auxiliary engines only. A condensing unit is also installed in the lower dynamo room for exclusive use of the seven dynamo engines.

For fire and bilge service there is, beside the eight auxiliary feed pumps, a pump in each engine room, $12 \times 10 \times 12$ inches, simple, vertical, double-acting. A crank-pit pump is attached at the forward end of each main engine to the crank-shaft end.

An ash-hoisting engine of usual type is fitted in the drum room over each boiler compartment. These engines are supplied with steam at reduced pressure.

A windlass engine is installed on the berth deck, forward. It is a simple, two-cylinder, vertical engine such as is commonly fitted on board U. S. naval vessels.

The evaporating plant consists of four horizontal evaporators of the type common to U. S. naval vessels, six distillers and necessary pumps. The evaporators have a combined capacity of 23,000 gallons and the distillers a combined capacity of 20,000 gallons in 24 hours. The pumps of this plant are an evaporator feed pump, $5\frac{1}{2} \times 5\frac{1}{2} \times 8$ inches; an auxiliary-feed pump $4\frac{1}{2} \times 4 \times 6$ inches; a fresh-water pump $5\frac{1}{2} \times 5\frac{1}{2} \times 8$

inches; and a distiller-circulating pump, also used as a sanitary pump, 12×16×18 inches.

An ice machine of the dense-air type is installed on the berth deck, forward, adjacent to the cold-storage rooms. Its capacity is three tons of ice per day, and it is connected for cooling the drinking water in the scuttlebutt, and the air in the magazines, in addition to its connections with the freezing tanks and the cold-storage rooms.

A steam-heating system, divided into circuits according to the needs of convenient heating, and controlled from the engine rooms, such as is usually fitted to vessels of the U. S. Navy, is installed.

The ship has a well-fitted machine shop capable of high-grade and varied work.

ELECTRIC PLANT.

This vessel and class are fitted with unusually large and well-equipped electric plants. The *California*, as have other vessels of the same class, has two dynamo rooms, under the protective deck, one above the other, immediately forward of and extending across the same width as the firerooms. There are seven generating units, built by the Union Iron Works, and a condensing system for the exclusive use of the engines of these units. In the upper room are three 50-kw. units and the switchboard of the system, and the lower room contains three 100-kw. units, one 50-kw. unit, and the condensing outfit.

The engines are designed for an initial working pressure of 150 pounds per gauge, and are vertical, cross-compound. The 50-kw. engines have cylinders of 10 and 16½ inches diameter, respectively, with 10-inch stroke, and the 100-kw. engines have cylinders of 13½ and 23 inches diameter, respectively, with 12-inch stroke. The generators supply a voltage of 125 at the terminals. There are supplied two spare armatures for each size of generator. The wiring is of the two-wire feeder system.

The generators supply current for the following uses:

a. Four motor generators for gun elevating, generators 0 to 125 volts.

b. One dynamotor, Holtzer-Cabot Co., size ½, for electric bells, which also have Leclanche cells, as a reserve battery.

c. One motor generator for telephones, same make and size as preceding. (Telephones work on 13 calling volts and 24 talking volts.)

d. Motors as below listed.

Quantity.	*Aggregate H.P.*	*Use.*
29	97.8	Stationary ventilating equipment.
6	...	Portable ventilating equipment.
46	4.3	Electric fans.
30	30	Operating watertight doors.
34	85	Chain-ammunition hoists.
1	1	Whip-ammunition hoist.
4	44	Turret-ammunition hoists.
4	14	Rammers, 8-inch guns.
2	100	Torpedo air compressor.
4	60	Turret turning.
4	14	Elevating 8-inch guns (supplied by motor generators "*a*").
8	200	Boat cranes.
7	210	Deck winches.
2	6	Fresh-water pumps.
1	1	Dish washer.
1	2.5	Dough-mixing machine.
1	7.5	Laundry.
2	10	Engineer's workshop.

e. Lighting fixtures, as below listed.

Quantity.	*Use.*	*Quantity.*	*Use.*
1	Top light.	294	Bulkhead lights.
1	Masthead light.	319	Steamtight lights.
1	Towing light.	69	Bunker lights.
2	Side lights.	152	Watertight portables.
15	Double brackets, living spaces.	6	Non-watertight portables.
1	Single bracket, living spaces.	72	Desk lights.
95	Ceiling fixtures.	6	Cargo reflectors.
126	Battle lanterns.	3	Binnacle lights.
36	Deck lanterns.	6	Engine-room telegraph lights.
33	Magazine lanterns.	2	Truck lights.
11	Signal lanterns.	6	Parabolic reflector search-lights.

The six searchlights require 450 ampères of current, and all other lights require a total of 625 ampères.

STANDARDIZATION RUNS.

On October 4th, 1906, the *California* left San Francisco for Santa Barbara Channel to undergo official trial runs, having had a builder's trial in San Francisco Bay, mainly for adjusting compasses for a sea trip, a few days before departure. The vessel reached Santa Barbara October 5th in a dense fog, and fog prevented making the standardization runs until October 9th.

For the purpose of establishing the relation between the revolutions of the main engines and the speed of the vessel, to be employed soon afterward in determining the vessel's speed from the engines' revolutions, the usual standardization runs were made over a straight course marked out parallel to the shore, just west of Santa Barbara harbor. Fourteen runs were made over this course, one sea mile in length, at revolutions varying from 128.41 starboard engine and 129.27 port engine, to 102.27 starboard engine 101.32 port engine, per minute, the runs alternating in opposite directions over the course to show influence of tide. From this data two speed curves were established, and a mean curve then determined, eliminating tidal effect. This curve is shown on the accompanying diagram, Plate I.

The greatest speed made during these runs was at the rate of 22.756 knots per hour, on a run to the westward, immediately following a speed at the rate of 22.514 knots per hour on the preceding eastward run.

Before starting on these runs, the vessel's draught was observed as follows:

Forward, feet and inches, ...	22– $8\frac{3}{4}$
Aft, feet and inches, ...	24– 7
Mean, feet and inches, ...	24– $1\frac{7}{8}$
Corresponding displacement, tons, . . .	13,780

On completion of the runs, the following draughts were noted:

Forward, feet and inches, ...	23– 7½
Aft, feet and inches, ...	24– 3¾
Mean, feet and inches, ...	23–11⅝
Corresponding displacement, tons, . . .	13,663

The estimated mean draught at the middle of the high-speed runs was 24 feet 1 inch, with a corresponding displacement of 13,750 tons.

Throughout these runs the boilers supplied more steam than the engines could use, the safety valves blowing almost continuously. The ship vibrated very little, and no vibration was noticeable on the upper deck over the engine and fireroom spaces at any speed.

The higher speeds caused heating of all crank-pin brasses to varying degrees, though none gave serious trouble except the I.P. brasses, due to the relatively great power developed by these cylinders. The trial was not interrupted, although these brasses pounded considerably when their white metal ran out. The heating of these brasses was due primarily to the fact that the crank pins were oiled by pipes from the crossheads along the connecting rods and into the crank-pin brasses, conveying oil to the brasses by gravity alone. The steady and sufficient delivery of oil by these pipes was greatly interrupted by centrifugal force at high speeds, depriving the pins of the necessary lubrication.

Completing the standardization runs, the vessel anchored again in Santa Barbara harbor, and spent the following day in fitting new crank-pin brasses where needed, and in otherwise making ready for the continuous four-hour run at 22 knots per hour, as required by contract. It was determined from the speed curve that the main engines should run at an average of 122.85 revolutions per minute to develop 22 knots.

FOUR-HOUR SPEED-TRIAL RUN.

On October 11th the speed run began in Santa Barbara Channel and continued about 2½ hours, when it had to be discontinued because of the lack of sufficient lubricant on the intermediate-pressure crank pins, causing them to heat excessively. On the following day the vessel returned to San Francisco, and, after refitting the crank-pin brasses where necessary, another attempt was made on October 22d to run the four-hour trial, this time at sea off San Francisco, though the same difficulty brought the run to an end after a continuance of about 2½ hours. A third attempt was made on October 26th, though this consisted only in getting the vessel up to full speed, when it was again demonstrated that the oiling pipes for the crank pins would not deliver oil satisfactorily at high engine speeds, although there was no difficulty with this lubrication when running at less than about 112 revolutions.

After this attempt, a centrifugal oiling arrangement was fitted to each main crank pin, which conveyed oil from a supplying receptacle to an annular grooved casting fastened to one web of each crank, concentric with the shaft. This casting was so fitted that centrifugal force impelled oil through a pipe from the casting into the hollow of the pin, and through the pin radially outward to the bearing surface. This simple apparatus solved the difficulty, and when, on November 12th, the vessel was taken to sea for another run, it was made successfully. No attempt was made to exceed the required engine speed, more than by a safe margin to insure the average required, as there was no premium for so doing; but the contract speed was made with ease, and it is apparent that after the engines have been in service sufficiently to work them to good bearing, this speed can be made with even greater ease.

On all these runs was used Harris Navigation (Cardiff) coal, screened and bagged.

There was installed in each fireroom an indicator to regulate the firing. This consisted essentially of a dial which flashed the numbers 1, 2, 3, 4, consecutively at any uniform

interval desired, at each flash ringing a small gong. All indicators were actuated electrically from a central source of power. In each fireroom the four furnace doors of the forward boiler were numbered alternately 1 and 3, and those of the after boiler were numbered alternately 2 and 4. The indicators were set to flash an identical number simultaneously in the four starboard firerooms, followed at the chosen firing interval by the same number simultaneously in the four port firerooms. In this way all the same numbered furnaces of the starboard boilers were fired at the same time, followed at the firing interval by all same numbered furnaces of the port boilers, this alternation between starboard and port continuing in cycles through the numbers in their regular order.

On the standardization runs the interval of firing was made twenty-four seconds between alternate starboard and port firings, but, as so frequent firing made much smoke and kept the safety valves lifted most of the time, the firing interval was lengthened finally to fifty-six seconds. Even at this interval on the successful four-hour run the boilers produced more steam than the engines needed to make contract speed, and very frequently a firing signal was passed without opening the furnaces in one or more firerooms, or at most opening only to level the fires, which were carried heavy at the back and light at the front, and were not sliced during the run. The boilers supplied steam so easily that the work of the firemen was light, and the air of the firerooms was very comfortable and not laden with dust. The smoke leaving the funnels was not heavy, even at its maximum.

The main engines, fitted with the centrifugal oiling gear for the crank pins, and all other machinery worked particularly well.

After this run the vessel's machinery was subjected to a thorough and careful inspection for defects developed by the tests imposed, but none were discovered. The only defect, that of the crank-pin oiling apparatus, had been entirely corrected by fitting centrifugal gear.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

NOVEMBER 12, 1906.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Mean steam pressure at boilers, pounds gauge	273.5	
Mean steam pressure, H.P. steam chest gauge, pounds	249.1	225.4
1st receiver (absolute), pounds	149.5	146.2
2d receiver (absolute), pounds	63.5	64.4
Vacuum in condensers, inches of mercury, mean	27.0	27.5

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	56.0	56.0
Discharge, degrees	119.0	100.0
Hotwell, degrees	122.0	122.0
Feed water, degrees	169.0	166.0
Engine room, upper platform, degrees	86.0	91.0
working platform, degrees	75.0	78.0
Firerooms, above grating, degrees	95.0	
working level, degrees	85.0	
Smoke stacks, average, degrees	620.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	124.17	124.99
Mean revolutions, both engines, per minute	124.58	
Pumps, main air	16.75	24.0
circulating	157.0	212.0
feed, d.s., per minute	25.6	22.8
fire and bilge	87.0	80.0
Blower engines	336.0	352.6
Speed of ship, in knots per hour	22.2	
Slip of propeller, in per cent. of its own speed, based on mean pitch	15.73	16.29
Air pressure in firerooms, in inches of water, mean	1.33	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	86.37	79.18
I.P. cylinder	46.68	51.31
F.L.P. cylinder	27.52	27.06
A.L.P. cylinder	26.88	27.24
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	55.63	56.53

INDICATED HORSEPOWER.

	Starboard.	Port.
Main engines, H.P. cylinder	2,952.8	2,725.1
I.P. cylinder	4,410.9	4,879.9
F.L.P. cylinder	3,534.7	3,503.2
A.L.P. cylinder	3,420.0	3,526.3
total	14,308.4	14,634.5
Collective H.P. of both main engines	**28,942.9**	
Air pumps, main	10.1	15.8
Circulating pumps, main	50.0	82.6
Feed pumps, main	160.0	119.2
Dynamo-condenser, air and circulating pumps	4.0	
Fire and bilge pumps	8.0	2.7
Forced-draft blowers	145.2	
Dynamo engines	114.4	
Total auxiliaries	**712.0**	
Collective I.H.P. main engines, air, circulating and feed pumps	**29,880.6**	
Collective, main and auxiliary engines in operation	**29,658.8**	

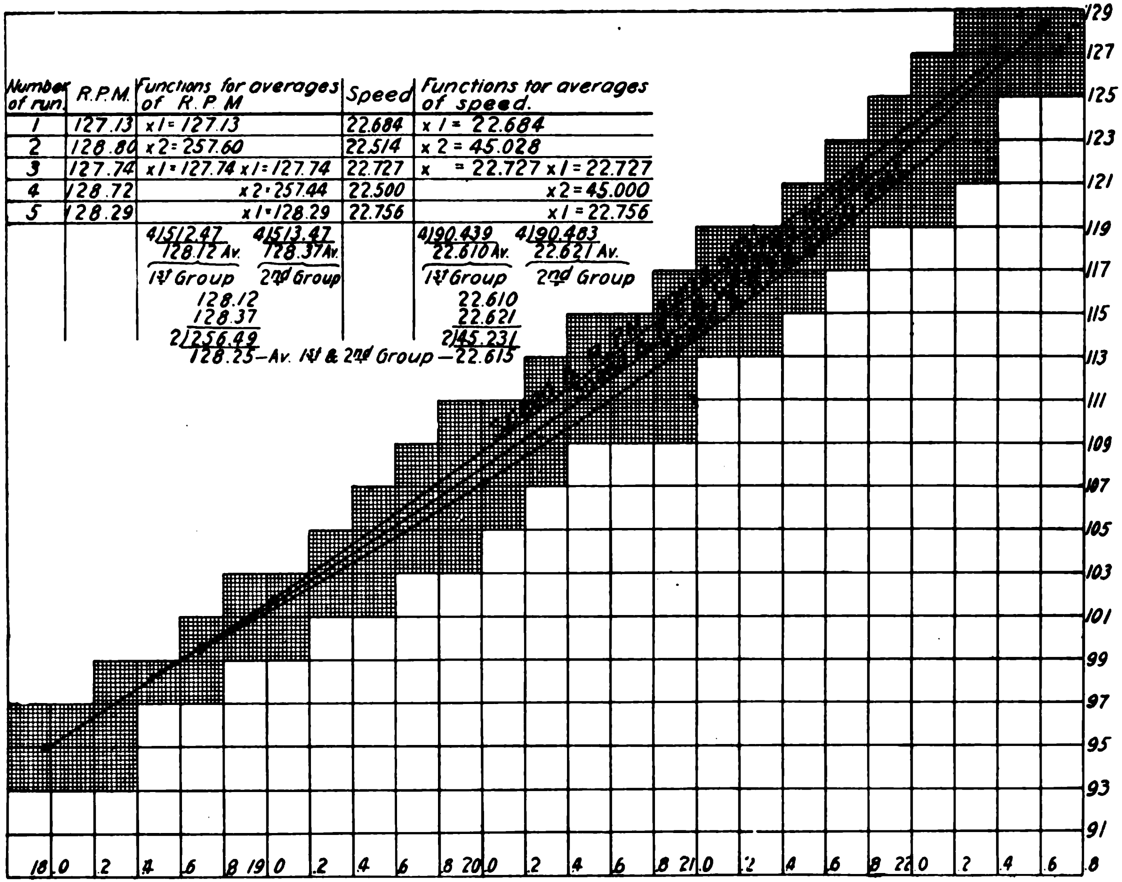
COAL.

Kind and quality used on trial: Harris Navigation, screened and bagged.

Pounds, per hour, main and auxiliary engines, during trial...... 47,190.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	18.63
main engines, air, circulating and feed pumps, per square foot of grate surface	18.46
main engines, air, circulating and feed pumps, per square foot of heating surface	0.414
Pounds of coal per I.H.P. per hour, collective, main engines, air, circulating and feed pumps	1.6
Pounds of coal per I.H.P. per hour, all machinery in operation	1.591
square foot of grate surface, per hour	29.65
heating surface, per hour	0.665
Cooling surface (main condenser), square feet per I.H.P. (total)	0.9698
Heating surface, square feet per I.H.P. (total)	2.39



Number of run	R.P.M.	Functions for averages of R.P.M.	Speed	Functions for averages of speed.
1	127.13	x 1 = 127.13	22.684	x 1 = 22.684
2	128.80	x 2 = 257.60	22.514	x 2 = 45.028
3	127.74	x 1 = 127.74 x 1 = 127.74	22.727	x = 22.727 x 1 = 22.727
4	128.72	x 2 = 257.44	22.500	x 2 = 45.000
5	128.29	x 1 = 128.29	22.756	x 1 = 22.756
		4)512.47 128.12 Av. 1st Group; 4)513.47 128.37 Av. 2nd Group		4)90.439 22.610 Av. 1st Group; 4)90.483 22.621 Av. 2nd Group
		128.12 128.37 2)256.49		22.610 22.621 2)45.231

128.25—Av. 1st & 2nd Group—22.615

"CALIFORNIA."—**Plate I.**

U. S. S. *SOUTH DAKOTA*.

DESCRIPTION OF OFFICIAL TRIALS.

BY J. J. RABY, LIEUTENANT, U. S. NAVY, MEMBER.

The *South Dakota* (Armored Cruiser No. 9), built by the Union Iron Works, San Francisco, California, is one of a class of six vessels authorized by Act of Congress dated June 7, 1900. She was the last of the class to have her official trials. Her sister ships are the *Colorado*, the *Pennsylvania*, the *West Virginia*, the *Maryland* and the *California*.

The contract for the *South Dakota* was signed by her builders on January 10, 1901, and she was to be completed in thirty-six months. Owing to labor troubles and the San Francisco fire the date of completion was extended in several stages to January 31, 1907, and the contractors have recently asked for a further extension to June 30, 1907. The contract price of the ship was $3,750,000, of which sum $1,500,000 was allotted to building and installing all machinery under the cognizance of the Bureau of Steam Engineering.

The keel was laid September 30, 1902, and the ship was launched July 21, 1904. Successful dock trials took place from March 20 to 27, 1906.

The *South Dakota* is practically a duplicate of the *California*, which vessel also was built by the Union Iron Works; and, as a description of the latter vessel by Lieutenant Danforth appears in this number of the JOURNAL, it is quite unnecessary to describe the *South Dakota* here, except to give the valve data for the main engines, which data will be found in Table I, and to give the power curves and the indicator cards for the auxiliary machinery, shown on Plate I to V.

Profiting by the experience of several unsuccessful trials of the *California* the contractors, before taking the *South Dakota*

out, fitted centrifugal oiling gear for the crank pins of the main engines. They also rebabbitted the crank-pin brasses. The original babbitt in these brasses was the regular navy composition, and the babbitt was laid in blocks in the brasses, as may be seen from the sketch on Plate VI. The extensive oil grooves took up much of what should have been bearing surface. The brasses as rebabbitted had oil grooves cut as per sketch on Plate VI, and the increase of bearing surface over that of the original brasses amounted to about 300 square inches for each complete brass. The metal used for rebabbitting is a special "Bearing Metal for Armature Liners" manufactured in San Francisco, and it is somewhat harder than the metal called for by the Bureau specifications. It may have been necessary to rebabbitt the brasses in order to get more bearing surface for the crank pins, but it is not believed that the metal substituted is superior to the regular navy composition, for it did not take on so good a polish, the metal appeared dull in color, and seemed to flake in places. The original brasses might have stood the test with the new oiling gear fitted, but the builders, in order to make certain of a successful trial, were taking no chances.

On December 18, 1906, the *South Dakota* left the works for her builders' trial, which took place in San Francisco Bay. After the compass was adjusted, the ship steamed around Angel Island for several hours, and the maximum speed during the run was 110 revolutions. With a few minor exceptions, the main engines, the boilers and the auxiliary machinery functioned satisfactorily. The next day the ship proceeded to Santa Barbara and anchored off that place during the forenoon of December 20th. On the run down the coast the engines were speeded up to 120 revolutions for a short time and everything worked smoothly.

STANDARDIZATION TRIAL.

On the morning of December 21st the *South Dakota* got under way shortly after 7·00, and at 8·14 began her standardization trial runs on the Goleta Point measured-mile course

situated about six miles west of Santa Barbara. The weather was fine and the sea smooth. Fourteen runs were made, seven to the westward and seven to the eastward, in order to eliminate influence of tide, and the first five runs were at maximum speed. The greatest horsepower developed on any run was 28,157.1 for main engines and 28,766.8 for all machinery, and the least was 9,262.5 for main engines and 9,507 for all machinery. The maximum number of revolutions per minute for a run was 125.58 starboard and 124.85 port, and the minimum 92.46 starboard and 91.87 port. The highest speed obtained on any one run was at the rate of 22.785 knots per hour. The H.P. and I.P. were set at maximum cut-off and the L.P. at minimum cut-off. The mean draught on trial was 24 feet 1 inch and the corresponding displacement 13,750 tons. The last run was completed at 11·20, after which the ship returned to her anchorage off Santa Barbara.

The standardization trial developed no serious defects. Engines, boilers and auxiliary machinery worked nicely, and though two valve guides ran hot and some of the crossheads and crank pins knocked slightly, there was no heating of bearings. Water was used freely on the bearings and about 380 gallons of lubricating oil was expended during the trial. So successful was the trial that the contractors' engineer did not deem it necessary to make any adjustments or inspect any bearing of the engines before the endurance run.

From the data obtained on the standardization runs a speed curve was constructed, which showed that the number of revolutions necessary to make the contract speed (22 knots) was 123.07. Plate VII shows the speed curve. On this plate is also shown the revolution-horsepower curve.

FOUR-HOUR FULL-POWER TRIAL.

On the morning of December 22, 1906, the *South Dakota* got under way and stood down the Santa Barbara Channel to the eastward for her four-hours' official trial. The weather was fine with light southerly airs and smooth sea. The trial began at 7·55 with the ship steering an easterly course. Dur-

ing the latter half of the run the course was changed to the westward up the Santa Barbara Channel. The trial was finished at 11·55. The mean draught for the trial was 24 feet 1 inch and the corresponding displacement 13,750 tons. The average revolutions per minute for the trial were 125.23, and the corresponding speed from the speed curve gives an average speed of 22.24 knots per hour for the four-hours' run. The cut-offs were set as for the standardization runs, H.P. and I.P. maximum, L.P. minimum. The main engines developed 28,158.96 I.H.P., and the total I.H.P. for all machinery was 28,842.81. The coal used was Harris navigation, good quality, hand picked and sacked.

The engines, boilers and auxiliary machinery worked exceedingly well during the trial. There were rather heavy knocks in S.A.L.P. crank pin and P.A.L.P. crosshead and crank pin, and slight knocks in I.P. and H.P. crank pins, but otherwise everything ran beautifully about the main engines, and an examination after the run showed that all bearings were cool. About 100 gallons oil per hour was used for lubrication during the trial, and water was used freely on the main bearings. The boilers steamed easily, and careful supervision of the firing, water tending, and running of forced-draft blowers kept the safety valves from lifting very much and the boilers from priming at all. The auxiliary machinery gave no trouble.

An electric time-firing device was used to give the signals for firing. The time was divided into eight firing intervals, and the average firing interval during the trial was about 56 seconds. The furnace doors on the forward boiler of each fire-room were numbered alternately 1 and 3, while the doors on the after boiler were numbered 2 and 4. The electrical connections in the time-firing device were so arranged that the signals for firing were made as follows: 1S, 1P, 2S, 2P, 3S, 3P, 4S, 4P. Each boiler was therefore fired about every 112 seconds on the average.

This same time-firing device was used for the signal to feed the boilers, and, except when necessary, each boiler was fed only during the time for firing that boiler. In this way the

work of the feed pumps was made uniform, the distribution of feed water was well regulated, and the water was kept at the proper level in the gauge glass. I believe this to be a very good way of feeding the boilers during full-power trials when the feed pumps are running at their greatest capacity, though when steaming at moderate speeds and the feed pumps are not hard worked the continuous feeding of the boilers might be more satisfactory.

The boilers steamed so easily to supply sufficient steam for the engines that the firing interval was rather long, and the average air pressure in the ash pans was only 1.35 inches, though 2 inches was allowed by the machinery specifications. The firemen seemed to be spending most of the time leaning on the handles of their shovels. The smoke pipes emitted little smoke and the average temperature of the smoke-pipe gases was 582 degrees at a point about 33 feet above the grates. The coal consumption was unusually low, for the pounds of coal per I.H.P. per hour amounted to only 1.46 for the main engines and 1.445 for all machinery. See page 26 for the data for the four-hour trial.

After the four-hour trial the ship had the usual maneuvering trials, but these were unsatisfactory for the reason that the main-engine throttle valves were of poor design and could not be readily closed. These valves were ordinary unbalanced conical valves worked by knuckle-joint gear, and were closed against the pressure in the steam pipe. The valves closed rather easily from full opening to the .3 mark, they were hard to close from .3 to .1, and to close the valve completely it took three strong men at the throttle-valve wheel with a 21-inch monkey wrench to increase the leverage. These throttle valves have been replaced by balanced valves, so that, as far as the throttle valves are concerned, no further trouble is expected in stopping and reversing the engines.

On the afternoon of December 22d the *South Dakota* got under way for San Francisco and anchored off the Union Iron Works the next morning. Altogether, the trials of the *South Dakota* were a great success. The ship was four days away

from the works of her builders, and in that time she steamed 780 miles from San Francisco to Santa Barbara and return, and performed most satisfactorily at both the standardization and four-hour full-power trials. This short time on a trial trip is an unusually fine record. Judging from the way she steamed and the way the machinery performed, it is safe to say that the *South Dakota* could have made much better speed had the engineer cared to drive her. A bonus for increased speed would have made her faster, and a forfeit for using water on the bearings might have kept it off.

The post trial inspection of the *South Dakota* at the Union Iron Works by the Engineering Trial Board showed only minor defects, the most serious of which (the main throttle valves) has already been noted. The main-engine cylinders, pistons and valves were in good condition, and the metallic packing of piston rods and valve stems was fine. The bearings were in very good condition, and only slight dragging of the babbitt was noticed in a few of the main bearings and crank-pin bearings, not enough to interfere with the free passage of oil through the oil grooves. The rod of the S.F.L.P. slide-valve balance piston was found to be broken. As this rod is one with the slide-valve stem a new stem has been fitted. The boiler tubes were found to contain some dirt and mill scale, but otherwise the boilers were in good shape. The auxiliary machinery was in excellent condition except that one of the cast-iron supporting brackets of the blower-engine cylinders over boiler Q was broken. The break has since been repaired.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

DECEMBER 22, 1906.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	278.4	
engines, pounds	259.2	259.7
H.P. steam-chest gauge, pounds	238.6	237.5
1st receiver (absolute), pounds	126.9	127.1
2d receiver (absolute), pounds	58.1	56.0
Vacuum in condensers, inches of mercury, mean	26.28	26.7

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Injection, degrees	58.1	60
Discharge, degrees	110.0	108.4
Hotwell, degrees	104.0	108.5
Feed water, degrees	168.7	181.2
Engine room, working platform, degrees	77.5	77.5
Firerooms, working level, degrees	88.7	88.7
Smoke stacks, average, degrees	582	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	125.58	124.88
Mean revolutions, both engines, per minute	125.23	
Pumps, main air	14.87	17.0
circulating	171.0	165.0
feed, d.s., per minute	24.0	23.0
fire and bilge	22.8	46.2
dynamo condenser, air and circulating	21.8	
Dynamos	350.0	
Blower engines	384.8	
Speed of ship, in knots per hour	22.24	
Slip of propeller, in per cent. of its own speed, based on mean pitch	16.53	16.07
Air pressure in firerooms, in inches of water, mean	1.35	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	94.94	96.88
I.P. cylinder	45.77	46.72
F.L.P. cylinder	25.9	24.6
A.L.P. cylinder	25.25	22.2
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	55.187	53.44

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	3,282.7	3,331.3
I.P. cylinder	4,373.91	4,439.4
F.L.P. cylinder	3,403.8	3,168.4
A.L.P. cylinder	3,284.45	2,875.0
total	14,344.86	13,814.1
Collective H.P. of both main engines	**28,158.96**	
Air pumps, main	9.5	11.0
Circulating pumps, main	56.0	54.0
Feed pumps, main	133.2	120.5
Dynamo-condenser, air and circulating pumps	5.2	
Fire and bilge pumps	1.7	4.0

Forced-draft blowers	181.0
Dynamo engines	101.0
Total	**677.1**
Collective I.H.P. main engines, air, circulating and feed pumps	**28,548.16**
Collective, main and auxiliary engines in operation	**28,842.81**

COAL.

Kind and quality used on trial..................Harris Navigation, good quality.
Pounds, per hour, main and auxiliary engines, during trial...... 42,180.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	18.027
main engines, air, circulating and feed pumps, per square foot of grate surface	17.84
main engines, air, circulating and feed pumps, per square foot of heating surface	0.402
Pounds of coal per I.H.P. per hour, collective, main engines, air, circulating and feed pumps	1.47
Pounds of coal per I.H.P. per hour, all machinery in operation	1.445
square foot of grate surface, per hour	26.36
heating surface, per hour	0.593
Cooling surface (main condenser), square feet per I.H.P. (total)	0.997
Heating surface, square feet per I.H.P. (total)	2.519

Table I.—U. S. S. *SOUTH DAKOTA*.—Valves and Valve Settings.

	H.P.		I.P.		F.L.P.		A.L.P.	
Number and type of valves	One-piston		Two-piston		One-d. p. slide		One-d. p. slide	
Diameter of valves, inches	$24\frac{1}{2}$		27		......		......	
Travel of valve, inches	......		......		......		......	
Side of valve on which steam is taken	Inside		Outside		Outside		Outside	
	Top.	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*
Width of port, and length of slide valve, inches	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$8-67\frac{1}{2}$	$8-67\frac{1}{2}$	$8-67\frac{1}{2}$	$8-67\frac{1}{2}$
† Steam opening, linear, inches	$3\frac{15}{16}$	$3\frac{13}{16}$	$2\frac{7}{8}$	$2\frac{15}{16}$	$2\frac{1}{8}$	$2\frac{9}{16}$	$2\frac{1}{4}$	$2\frac{1}{8}$
Steam opening, area of, in square inches	266.7	258.2	435.0	444.6	285.2	344.3	302.1	335.8
† Exhaust opening, linear, inches	$3\frac{13}{16}$	4	Full	$4\frac{1}{8}$	Full	Full	Full	Full
Exhaust opening, area of, in square inches	258.2	270.8	680.5	624.6	536.6	536.6	536.6	536.6
† Steam lap, inches	$1\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{9}{32}$	$2\frac{1}{8}$	$2\frac{3}{4}$	$2\frac{11}{16}$	$2\frac{3}{4}$	$2\frac{9}{16}$
† Exhaust lap, inches	$1\frac{3}{16}$	$1\frac{1}{4}$	$\frac{17}{32}$	$\frac{31}{32}$	$-\frac{1}{4}$	$\frac{7}{16}$	$-\frac{1}{4}$	$\frac{7}{16}$
† Steam lead, linear, inches	$\frac{11}{16}$	$\frac{15}{16}$	1	$\frac{29}{32}$	$\frac{13}{16}$	1	$\frac{13}{16}$	$\frac{15}{16}$
† Cut-off in decimal of stroke, Maximum	.909	.875	.792	.674	.661	.565	.667	.573
† Cut-off in decimal of stroke, Minimum	.725	.625	.552	.419	.422	.338	.445	.344

† To be taken from finished work.

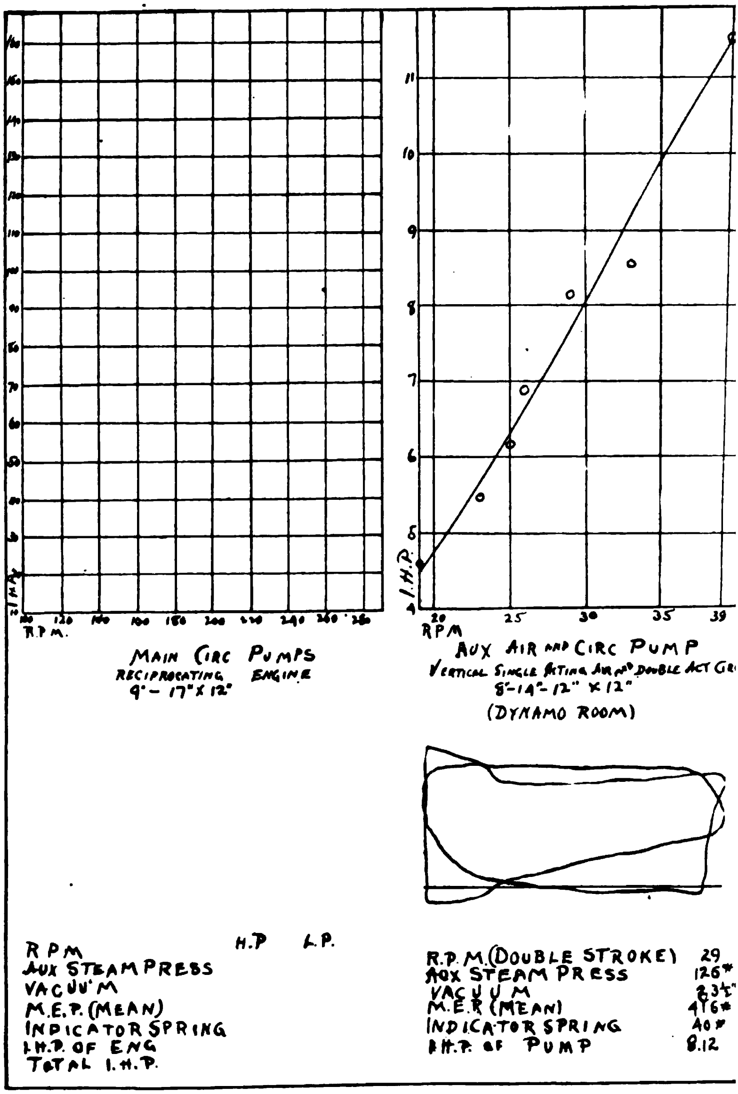

I.H.P.
R.P.M.
MAIN CIRC PUMPS
RECIPROCATING ENGINE
9" – 17" X 12"
I.H.P.
RPM
AUX AIR AND CIRC PUMP
VERTICAL SINGLE ACTING AIR AND DOUBLE ACT CI
8"–14"–12" X 12"
(DYNAMO ROOM)
RPM
AUX STEAM PRESS
VACUU'M
M.E.P. (MEAN)
INDICATOR SPRING
I.H.P. OF ENG
TOTAL I.H.P.
H.P
L.P.
R.P.M. (DOUBLE STROKE) 29
AUX STEAM PRESS 126#
VACUUM 23½"
M.E.P. (MEAN) 4.6#
INDICATOR SPRING 40#
I.H.P. OF PUMP 8.12

Plate I.

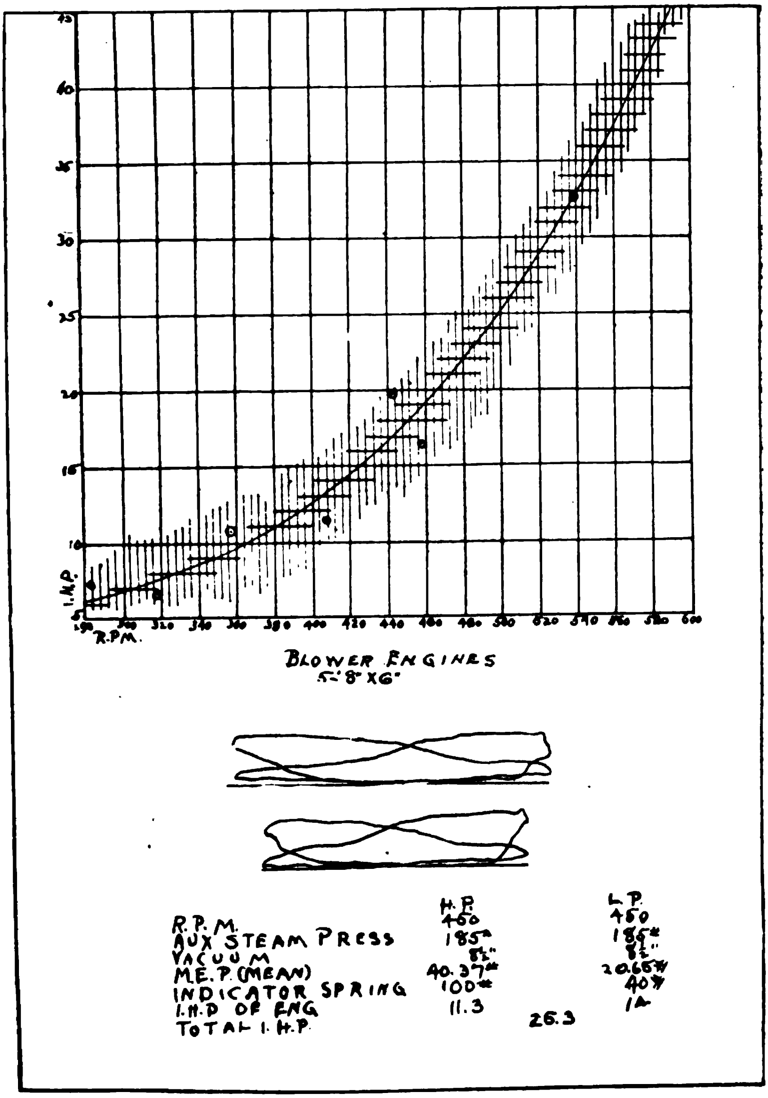

I.H.P.
R.P.M.
BLOWER ENGINES
5"-8"X6"
H.P.
L.P.
R.P.M. 450 450
AUX STEAM PRESS 185# 185#
VACUUM 8½" 8½"
M.E.P. (MEAN) 40.37# 20.65#
INDICATOR SPRING 100# 40#
I.H.P OF ENG 11.3 14
TOTAL I.H.P 25.3

Plate II.

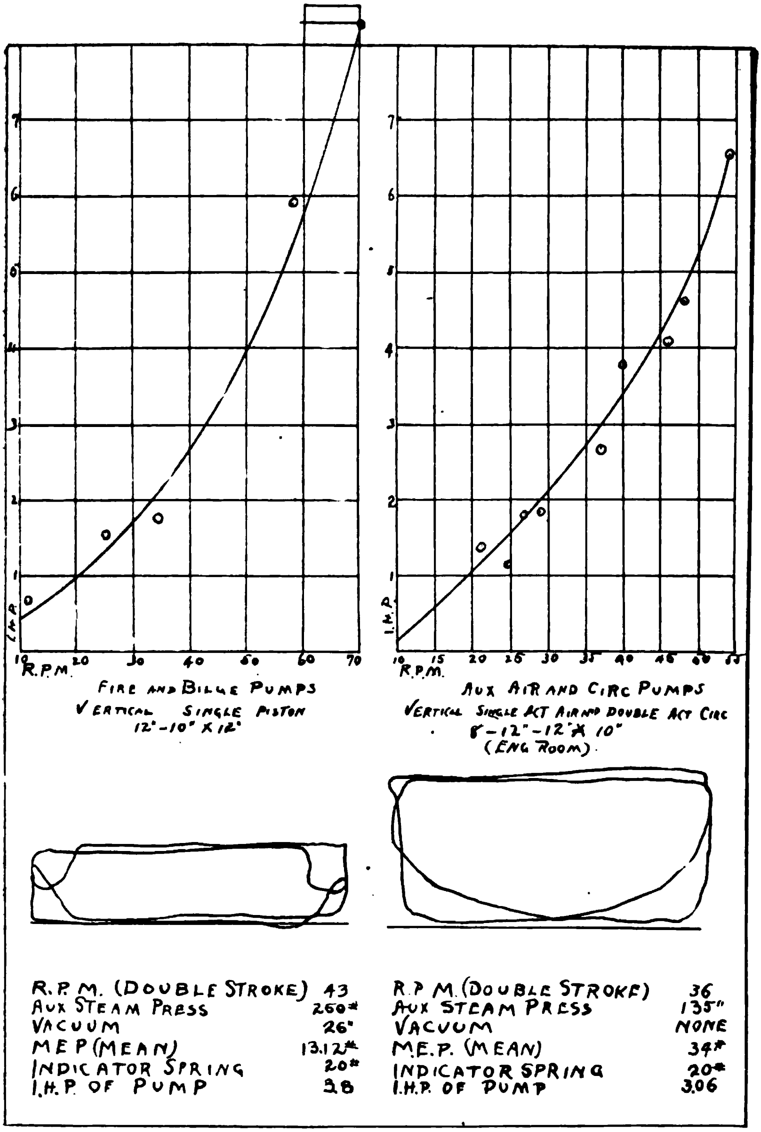
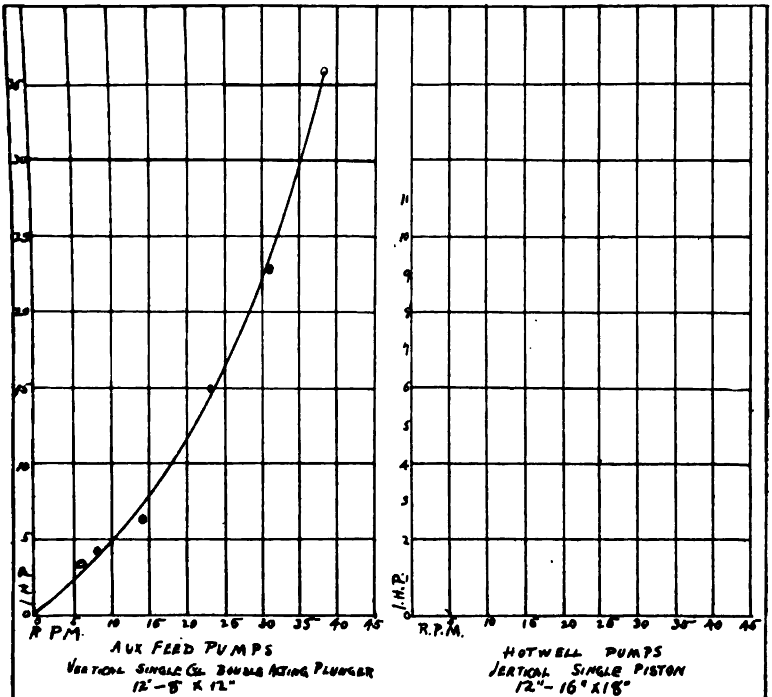
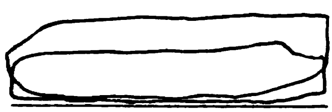

I.H.P.
R.P.M.
FIRE AND BILGE PUMPS
VERTICAL SINGLE PISTON
12"-10" X 12"
AUX AIR AND CIRC PUMPS
VERTICAL SINGLE ACT AIR AND DOUBLE ACT CIRC
8"-12"-12" X 10"
(ENG ROOM)
R.P.M. (DOUBLE STROKE) 43
AUX STEAM PRESS 260#
VACUUM 26"
MEP (MEAN) 13.12#
INDICATOR SPRING 20#
I.H.P OF PUMP 3.8
R.P.M. (DOUBLE STROKE) 36
AUX STEAM PRESS 135"
VACUUM NONE
M.E.P. (MEAN) 34#
INDICATOR SPRING 20#
I.H.P. OF PUMP 3.06

Plate III.





RPM (DOUBLE STROKE)	23	R.P.M. (DOUBLE STROKE)
AUX STEAM PRESS	230#	AUX STEAM PRESS
VACUUM	21"	VACUUM
M.E.P. (MEAN)	96#	M.E.P. (MEAN)
INDICATOR SPRING	150#	INDICATOR SPRING
I H P OF PUMP	149	I.H.P. OF PUMP

Plate IV.

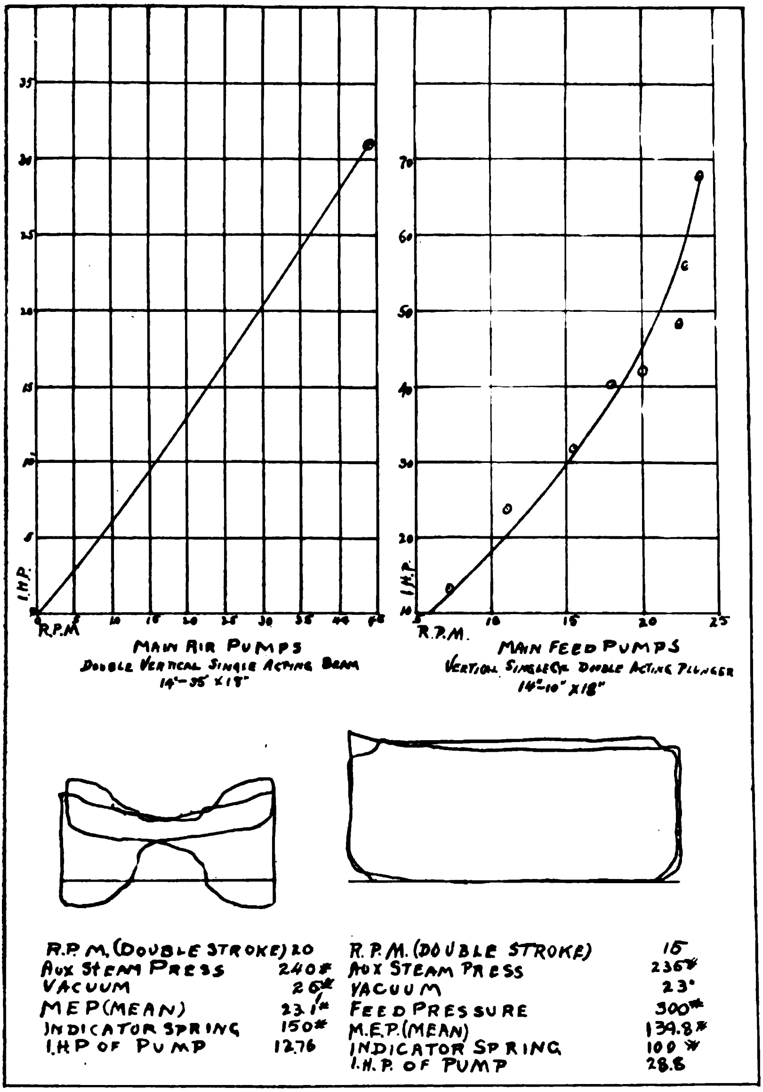

I.H.P.
R.P.M.
Main Air Pumps
Double Vertical Single Acting Beam
14"-35" x 18"
I.H.P.
R.P.M.
Main Feed Pumps
Vertical Single Cyl. Double Acting Plunger
14"-10" x 18"
R.P.M. (Double Stroke) 20
Aux Steam Press 240#
Vacuum 26"
MEP (Mean) 23.1#
Indicator Spring 150#
I.H.P of Pump 12.76
R.P.M. (Double Stroke) 15
Aux Steam Press 236#
Vacuum 23"
Feed Pressure 300#
M.E.P. (Mean) 134.8#
Indicator Spring 100#
I.H.P. of Pump 28.8

Plate V.

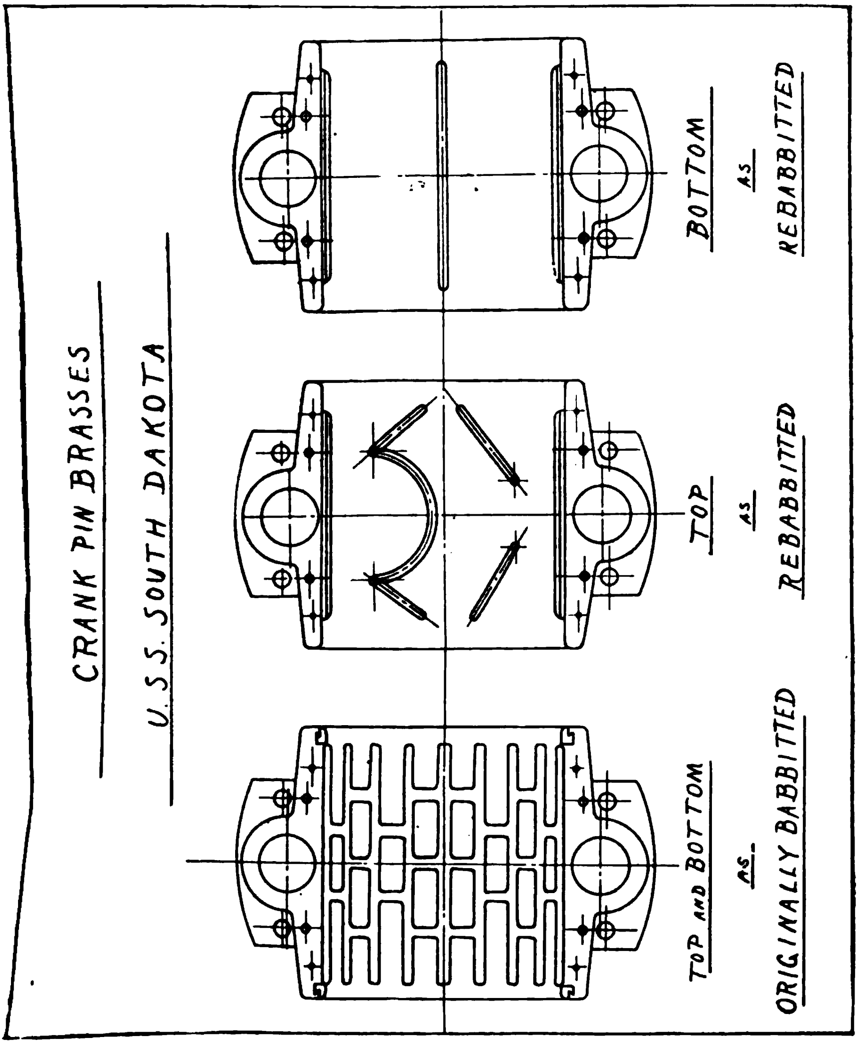
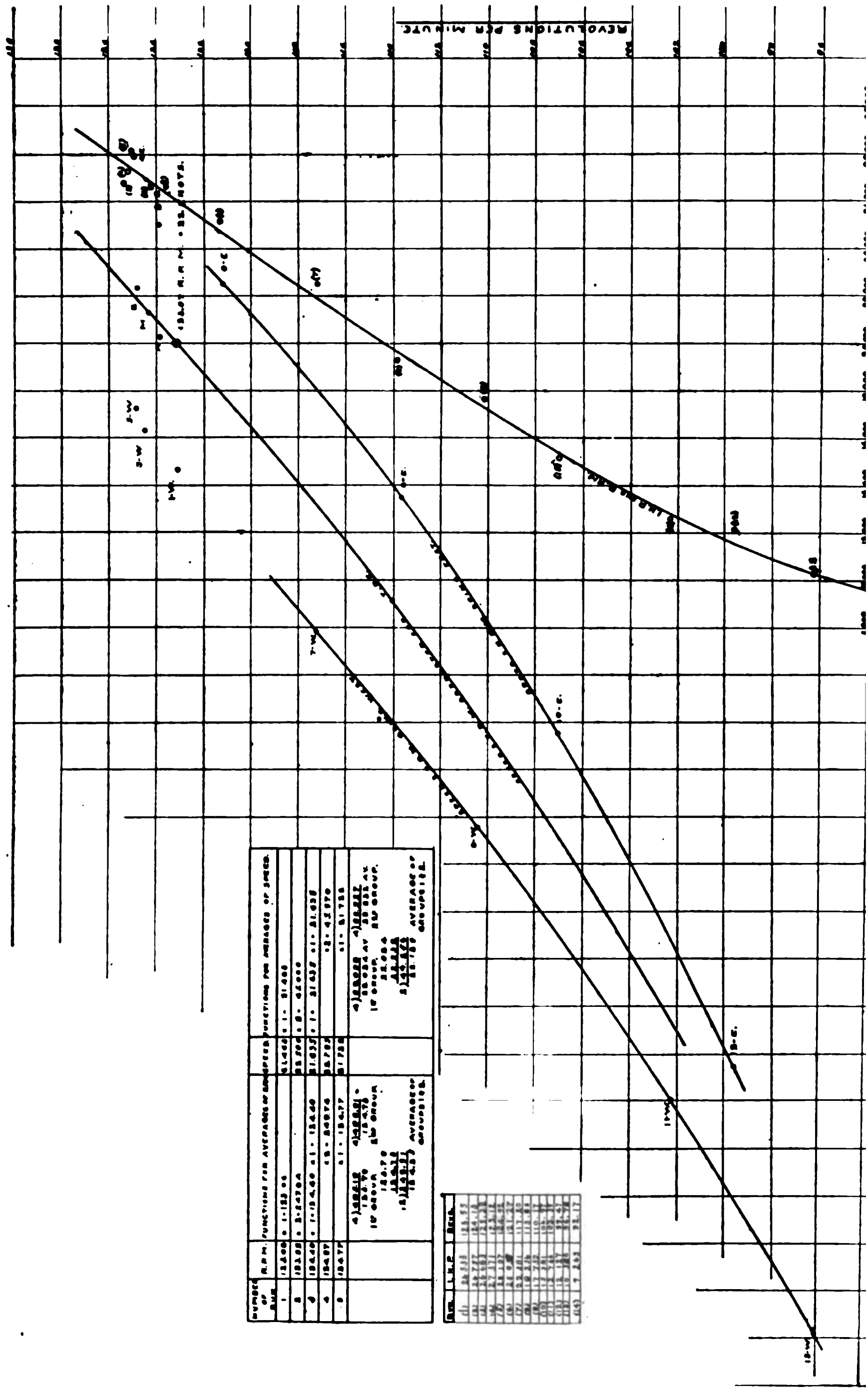
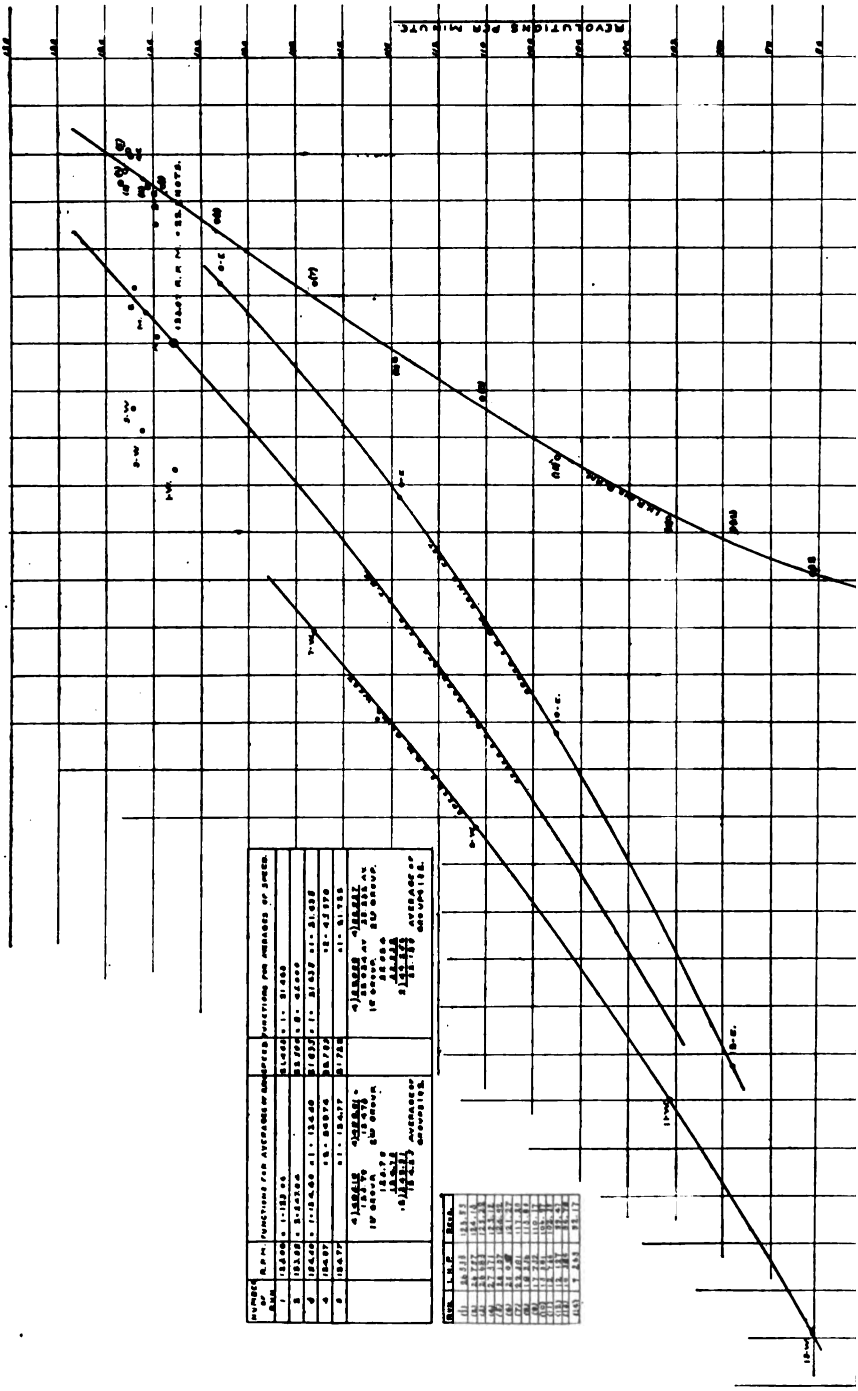
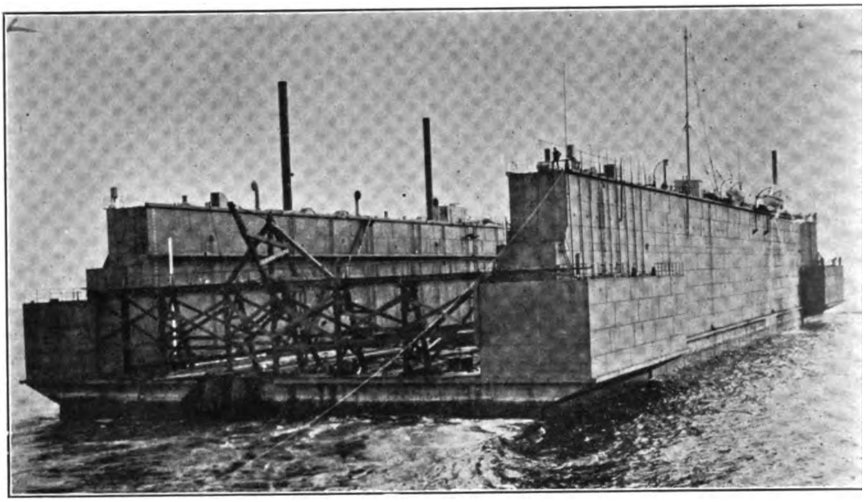

CRANK PIN BRASSES
U.S.S. SOUTH DAKOTA
TOP AND BOTTOM
AS
ORIGINALLY BABBITTED
TOP
AS
REBABBITTED
BOTTOM
AS
REBABBITTED

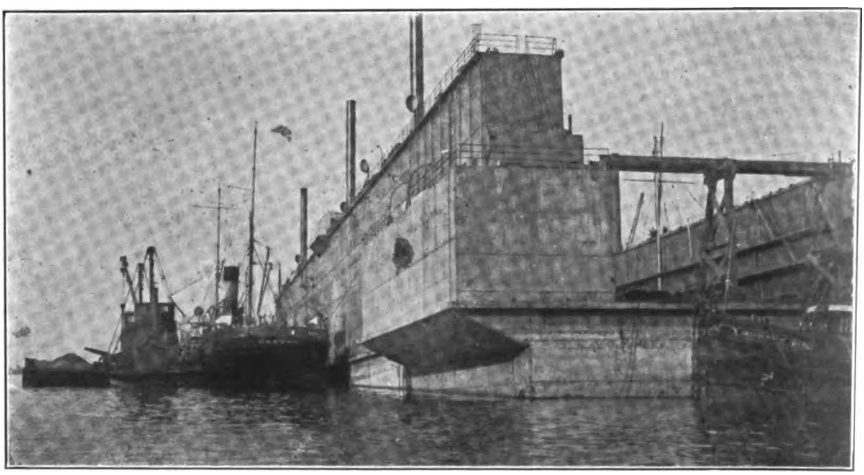
Plate VI.


REVOLUTIONS PER MINUTE


REVOLUTIONS PER MINUTE.

Cavite Dry Dock.





Cavite Dry Dock.

THE CAVITE DRY DOCK AT SEA.

BY COMMANDER F. M. BENNETT, U. S. N. MEMBER.

The steel floating dry dock built at Baltimore for use in the Philippine Islands has been fully described by articles in previous issues of this JOURNAL, from which articles I borrow the name, though I think it a misfit, because the water is so shoal in Manila Bay that this dock could not be used anywhere in the immediate vicinity of the U. S. Naval Station at Cavite. The act of Congress authorizing its construction is, I believe, the authority for fixing the name Cavite upon it. When launched, the dock was christened *Dewey*, in honor of the Admiral of the Navy, but in long association with it we of the towing squadron always found it awkward to apply that distinguished name to it, or, indeed, to use any name for such a remarkable object, and in daily conversation "it" and "the dock" were the more usual and milder terms by which it was designated.

The last chapter in the history of this dock that has appeared in the pages of the JOURNAL was, I think, the article by a member describing the self-docking tests that were successfully carried out during the month of July, 1905, in the Patuxent river. Beginning from that point, I shall endeavor to describe briefly some of the engineering problems that arose while the dock was in transit from the United States to the Far East. Preliminary to such description, however, it is necessary to give some account of the special equipment of the dock and the towing ships to fit them for the work they had before them. At the conclusion of the self-docking tests the dock was pronounced a complete success, and so it probably was if it could have remained at its moorings in the Patuxent river and been used there for docking ships. As a

sea-going structure it was at that time far from complete, and as the long sea voyage was a compulsory service at the beginning of its career, its success at the time referred to cannot be regarded as complete when provision for that first service had been overlooked. Such important essentials as anchors and chains and a capstan for handling them, life boats, and running lights for a vessel underway, had to be provided, besides many minor but necessary articles and fittings before the dock could be safely moved. When the builders' men left the dock, only a few days before we intended taking it to sea, it was discovered that the shovels in use in the fireroom actually belonged to them, and that none had been provided for the dock, which incident may give an idea of the many little omissions that came to light to distract us during the hurried work of preparation for the long sea voyage.

The shape of the dock was very unfavorable for towing, the submerged section being rectangular and presenting a square, wall-like surface from seven to ten feet in depth and 135 feet wide to be dragged bodily through the water. In good, smooth weather and without aid from wind or currents we found four and one-half knots to be about the maximum speed we could make with the combined power of the towing ships, which speed also was about the limit that the towing gear could stand. More powerful ships could have pulled more, of course, but any rise in speed would have been met by such increased resistance from the dock that very much stronger towing lines than any we had would have been necessary. In one of the articles published in the JOURNAL there were drawings showing several designs of floating docks that had been submitted in competition when proposals for building this one were invited. At least one of these, if I remember correctly, showed a dock with the ends pointed, to facilitate its being towed at sea, and it seems, in view of the long journey ahead of the Cavite dock, that some such feature should have been included in its design. The large Havana dock, I have been told, was provided with a false, pointed

bow when it was towed out, and also with an enormous rudder, both of much value in keeping it manageable.

Our greatest troubles in the towing expedition may be directly charged against the unyielding opposition to progress of the square-fronted dock; it was a sufficiently ugly thing to tow in good weather, but in high head winds and seas, of which we encountered more than our share, it frequently took charge beyond any power to control. On such occasions the seas and the dock gave a very good presentation of that cataclysm upon which young engineers frequently speculate, which is supposed to result from the collision of an irresistible force with an immovable object, the dock playing the part of the immovable object with temporary success but with ultimate failure in all cases. Broken tow lines, the dock adrift, and a great amount of time and labor demanded to get hold of it again were the results. On some, if not all, of these occasions of disaster I am sure we could have retained hold of the dock had it been provided with a sea-going bow, but as it was, the great seas beating upon the square front of the dock drove it backward and put a strain upon our towing gear that could not be withstood. A pointed bow, also, would have enabled us to make the voyage to the Philippines at considerably greater speed, with corresponding reductions in time, and in cost of coal and towing gear consumed.

The vessels selected for the expedition were the supply ship *Glacier*, the naval colliers *Brutus* and *Caesar* and the tug *Potomac.* The *Glacier* and *Potomac* were manned by naval officers and crews and the colliers by merchant officers and crews. The principal dimensions of these vessels are: *Glacier*, 7,000 tons displacement, 353 feet length, 46 feet beam, 25 feet draught, 4,000 horsepower, 12.5 knots speed; *Brutus*, 6,600 tons displacement, 329 feet length, 41 feet beam, 23 feet draught, 1,200 horsepower, 10 knots speed; *Caesar*, 5,000 tons displacement, 322 feet long, 44 feet beam, 21 feet draught, 1,500 horsepower, 10 knots speed; *Potomac*, 785 tons displacement, 139 feet long, 28 feet beam, 12 feet draught, 2,000 horsepower, 16 knots speed. In actual towing in aver-

age weather the *Glacier* developed (by indicator) about 1,700 horsepower, the *Brutus* 1,100 and the *Caesar* 1,200, or 4,000 in the aggregate for the three ships, the combined displacements of which were more than 18,000 tons. This seems small power to move so much weight and have anything left to apply to towing a 12,000-ton dry dock, and so it was; but, all things considered, the vessels were better suited for the work than higher-powered cruisers or battleships would have been. Capacity for carrying coal for long periods at sea was a consideration that led to the selection of the colliers, and the demand for provisions for the expedition for a long period selected the *Glacier*, which ship has cold-storage space for 1,000 tons of frozen meats. The fuel question was met in her case by filling her cargo holds and part of the cold-storage space with 2,000 tons of coal. Besides providing cargo coal to burn when the bunker supply ran low, these cargoes were necessary with all the ships to keep them deep in the water and their screws submerged for effective operation. The towing ships were all single-screw vessels, which was their weakest point, as they were clumsy and hard to manage at times when handiness would have been a great comfort.

Except the *Potomac*, none of these ships had any special appliances for towing, and all had therefore to be prepared after their selection for the expedition. The chief item in this preparation was the installation of a towing machine on each vessel. The machines obtained were those made by the American Ship Windlass Company of Providence, R. I., and were the size designated as No. 5, the largest that is made except on special orders. This machine is a two-cylinder steam engine, the crank shaft of which, through a pinion and drum gear, drives a large drum upon which the wire tow line, two inches in diameter and 1,200 feet long, is reeled. The ratio of the gearing is 5 to 1; the steam cylinders are 16 inches in diameter and 16 inches stroke of piston. Steam is always on this engine when towing, being admitted through a partly-open regulating valve, and its distinctive feature is in providing this steam cushion to take up any sudden jerks on the

tow line. When any increased strain comes on the line, it is drawn out against the pressure of the steam in the cylinders, an automatic attachment at the same time screwing open the regulating valve until a sufficient increased pressure is admitted to the cylinders to balance the strain on the line. Then, when the strain becomes normal again, the machine slowly winds in the few fathoms of wire that ran out, the automatic gear closing the regulating valve at the same time until equilibrium between steam pressure and strain on the line is re-established. So well did the machine work that a mark on the line always came back to within a few inches of its original place. In rough weather, with the ships pitching and wallowing deeply and the strain on the tow line consequently very variable, the machine would be working constantly, paying out wire violently and reeling it back gently, but in fairly smooth weather it would sometimes stand for hours at a time without a movement. A by-pass valve and operating lever are provided for working the machine independently of the automatic gear.

Like most vessels built for commercial purposes, the towing ships all had the hand wheel and hand-steering gear on the upper deck as far aft as the taffrail, and these were directly in the way of the tow lines. On board the colliers the difficulty was met by shifting the wheel to the deck below, connecting it to its former spindle by sprocket wheels and chain, and by building heavy timber casemates over the gear remaining on the upper deck. On board the *Glacier* two deck houses far aft made the problem more difficult, and it was solved eventually by erecting across the after deck three heavy steel-and-oak arches over which the tow line was led clear of the houses and the hand wheel. This was not a good arrangement, as the wire hawser had to lead up from the towing drum to pass over the arches, and, of course, down again toward the water when it left the aftermost arch. With the great strain upon it, the wire would cut into the hard oak tops of the arches as it worked back and forth, and kept us busy patching and repairing them. There was, fortunately, a quantity of quar-

ter-inch sheet copper on board, and we discovered that that made an excellent covering for the arches in the wake of the tow line; it would get cut through in the course of a week or so, but that was a great improvement over having to piece and patch on the frames every day. I never knew before how tough so soft an appearing metal as copper is, as it certainly stood more punishment from the tow line than could iron or steel plates of the same thickness. If any scientific member of this Society doubts this statement I invite him to attack a soft piece of copper with a file or with hammer and chisel and he will learn something new.

Twelve steel hawsers, of the size denominated six-inch, but rather more than two inches in diameter, were provided for use with these machines. They were each 200 fathoms (1,200 feet) long, weighed about 7,000 pounds each, and were fitted with a thimble and shackle in one end, the other end being pointed to pass through a 2-inch hole in the flange of the towing drum, where it was secured by clamps. Four of these were given to each of the three towing ships. Big as they were, they were found insufficient for the heavy strains that mid-winter Atlantic weather put upon them, and at the first opportunity 8-inch Bullivant steel hawsers, obtained from England, were substituted for them. Four of the 6-inch wires were made by the Roebling Company in the United States; the other eight came from England.

Twelve 15-inch manila hawsers were also provided. These were 200 fathoms long and were very big, handsome lines when new, weighing five tons each. Five of them were doubled by joining the ends and fitting thimbles in the bights thus formed, the two parts of the line being stopped together every two fathoms the whole length by stout seizings of ratline stuff. As thus made up, each of these spans with its thimbles, shackles and seizings weighed over six tons, and was, as may be readily understood, an awkward thing to handle, particularly after it had been in use for towing and had become thoroughly waterlogged. The other seven 15-inch lines were left single, with an eye for a towing thimble in one

end only. The shackles, thimbles and other fittings for all these lines were made in the equipment shops at the Boston Navy Yard, and were enormous for such things, as may be realized by looking at Plates 1 and 2 accompanying this article. In distributing these manila towing spans for use, two of the double and one of the single ones were put on the dock, and one double and two single spans on each of the three ships. The dock was provided with four bridles, each 90 fathoms long, made of 2½-inch chain cable, the legs securely bitted at opposite corners of the dock, and a towing span shackled in the middle where the legs joined. Two bridles were kept in place at each end of the dock, and at the end not in use (the stern for the time being) a towing span was kept shackled to the bridle and ranged ready for running at any time; this to save time in taking the dock in tow after a breakdown. We towed the dock from either end indiscriminately, whichever one was the easiest to get hold of under the existing circumstances. To make this possible, a wooden trestlework-bridge and derrick for handling the heavy bridles and towing spans had been built at the stern end of the dock as an afterthought while we were preparing to take it to sea.

The expedition started from the Patuxent River December 28, 1905, and passed out at the Capes of the Chesapeake about 10·30 P. M. the 29th. The *Brutus* was next the dock and had two of the double manila spans, shackled end to end, between the 6-inch wire on her towing drum and the bridle on the dock. The *Caesar* was next ahead of the *Brutus*, with one double 15-inch span shackled into the end of the wire on her drum, the other end being fast on the bow of the *Brutus* by a chain bridle dipped through its shackle. In getting underway, and for several days thereafter, the *Potomac* towed ahead of the *Caesar*, but her coal capacity did not permit of continuous work of this kind, and the permanent arrangement was with the *Glacier* ahead of the *Caesar*, the *Potomac* being in tow herself astern of the dock. For several thousand miles at first the *Glacier* towed with one of the long 15-inch

single manila spans between her 6-inch wire and the bow of the *Caesar*, but after that had been carried away twice and cut several times in emergencies she used a double span, the same as the other ships. In open water all the ships kept nearly all of their 6-inch wire hawsers unreeled from the towing engine, which made the tow considerably more than a mile long from the bow of the *Glacier* to the stern of the *Potomac*.

The first check from bad weather occurred January 4, when we were only about 500 miles along on our journey. A moderate gale from S.S.W. and heavy sea prevailed at the time. The two colliers were fast to the dock and were facing the weather, steaming ahead only enough to keep steerage way, but their violent plunging combined with the beating of the heavy seas against the dock distressed the towing engines greatly, notwithstanding the long tow-lines in use. About the middle of the afternoon the *Caesar's* machine was disabled, several teeth breaking out of the drum gear and catching between the gear and the pinion on the crank shaft where they caused the crank shaft to be lifted out of its bearings, breaking the caps, and, of course, putting the machine out of action. They managed to bitt the line, and thus held on without losing it, and always thereafter the *Caesar* towed with her line bitted, being convinced that the towing machine was inadequate to the work of our expedition. The drum gear was temporarily repaired by putting in studs in place of the broken teeth and thus made serviceable for paying out or reeling in slack wire. The teeth of the drum gear were not machined parallel, but were slightly tapered lengthwise, just as the pattern had been made, to permit of its being drawn out of the sand in molding. This was their source of weakness, as they engaged chiefly at the thicker ends, and therefore brought on an unequal strain that caused those ends to chip and break; this happened to the towing machines on all three ships to a greater or less extent and led to the complete disablement of the machines of the *Brutus* and *Caesar*.

About a week later, January 12, the dock broke adrift from us for the first time. The wind was only moderate, but a

very heavy sea had been running all day, causing the ships to roll abominably, and no good cause for the towing gear to fail existed; but late that night the 6-inch wire line in the train between the *Brutus* and the dock parted and let the dock go adrift. I shall not in this paper attempt to describe the toil and anxiety that such an accident brought with it, but they may be readily imagined by anyone reflecting upon the mass and weight of wet lines to be hauled on board, the clearing and preparation of them for use again, the handling of heavy shackles in the dark, and then the task, sometimes really dangerous, of getting hold of the drifting and unmanageable dock. In the whole history of the expedition the dock was adrift six times—four times in rough seas, once in a calm when there was absolutely no provocation for it, and once as an incident of shortening the tow-lines preparatory to entering port. The parts of the towing gear that failed and thus permitted these accidents were the 6-inch steel hawsers, twice; a double 15-inch manila span, once; and once each for a 2-inch chain bridle, a double shackle and a triangular shackle. Only a few days after we had resumed towing after the breakdown above mentioned, the single 15-inch manila line between the *Glacier* and *Caesar* parted, but we did not reckon that a calamity, because two ships remained towing the dock and the general progress was not stopped while the *Glacier* repaired damages. The same thing happened again at a later period in the voyage, after which the use of the single 15-inch lines was discontinued.

Beginning January 24 we had a period of almost a month of head winds and discouraging conditions generally. We were about the middle of the Atlantic Ocean trying to make a course east (true) along the 28th parallel of latitude, and met the wind from N.E., East, and S.E., almost without change and of such force as to impede us greatly, while frequent gales gave us a vast amount of hard labor and trouble. Daily "runs" of less than forty miles became common, and one day the noon position by observation put us twenty-four miles west, or astern, of the place we had been in the previous

noon. January 25, while facing a moderate to fresh gale from S.E., the *Glacier* had to cut off from the tow to reduce the strain on the lines of the other ships, but that night the great strain and heavy sea overcame the towing machine on the *Brutus*, which was totally disabled. Teeth were stripped off the drum gear around almost half its circumference, the crank shaft was thrown out of its bearings and bent, both connecting rods badly bent, and other parts either broken or bent beyond repair; the 6-inch wire line parted near the towing drum at the same time, and of course the dock was adrift. The weather continued so bad that it was forty-nine hours before we got the dock in tow again, during which time it had drifted about 100 miles back toward the United States. Only twenty-nine hours later it broke adrift again by the parting of a 2-inch chain bridle over the stern of the *Brutus*, and was at large a day and a half before lines were gotten to it and the towing hawsers hauled over ready to use.

In the accidents that had happened thus far the longest pieces of the broken 6-inch wire lines had been hauled on board the dock, and these were spliced together, making one span about 250 fathoms long, which was put into the place of a damaged 100-fathom 15-inch manila span between the *Brutus* and the dock, the forward end of it being shackled into a new 6-inch wire line on the *Brutus*. This line, owing to the *Brutus*' towing machine being disabled, was nearly all paid out, the inboard end being bitted and the bitts backed to the mainmast by chain cables taken from the anchors. There was thus a clear run of tow lines of over 3,200 feet, or more than half a mile, from the *Brutus* to the dock, the great length and weight of which was its own protection, and there were no more breakdowns as long as it was in use. As made up this time, the whole length of the tow from the *Glacier* to the *Potomac* was more than a mile and a half.

While toiling forward against head wind and seas with enough troubles of her own, the exigency of the *Potomac* in regard to coal began to force uncomfortable attention. As before stated, that vessel was usually towed, but in very rough

weather she had to cast off and take care of herself, and consequently had to keep enough coal on hand to enable her to weather a gale, besides sufficing for her daily needs. We watched her noon reports showing the steady diminution of her supply from day to day and hoped fervently for a let-up in the merciless wind that would make it possible for her to come alongside the *Glacier;* but no such abatement came nor seemed probable, until finally, when the situation forbade any more waiting, the *Glacier* cast off from the head of the line and devoted herself to the *Potomac* for the next six days. At first we took her alongside and gave her fifteen tons of coal in about half an hour, but then had to desist, as the sea was so heavy that both vessels were sustaining serious damages. Nearly a day was spent in trying to put into operation an overhead wire transporter, but the violent motion of the ships, the pitching of the *Potomac* especially, rendered this scheme entirely impracticable. Then, with the *Potomac* in tow, we got an under-water line in operation and began sending her coal in bags, hauled through the water. That was slow at first, and during two days we gave her only about as much as she burned, but the next day, with experience and with much heavier gear to work with, we gave her four days' supply. The next morning the weather was moderate for a few hours, which we took advantage of to get her alongside, and gave her enough to make her safe for two weeks longer. It was a hard and trying week, but valuable in showing that men can do almost anything when they have to.

The dock had now become a source of anxiety because the people on board it thought it threatened to fall apart. The end pontoons were secured to the overhanging side walls of the central pontoon by a system of plates, angle irons and bolts, and weakness had developed in the riveted angle irons that secured the vertical fastening plates to the deck of the end pontoons. The dock was very buoyant and always had considerable motion, both rolling and pitching, in a seaway. This brought a constantly varying series of stresses and strains upon the riveted joints spoken of, which being continued for

weeks without rest as the dock pitched about in the rough sea began to show disastrous results. Many of the rivets were worked loose (6,000 out of 7,000 had to be renewed), and this loosened the angle irons so that they had considerable play as the ends of the dock rose and fell, the mischief, of course, multiplying as the amount of loose motion increased. There was yet strength enough in the joints to resist considerable bad weather, but it was not prudent to let the injuries go on increasing, and the commander of the expedition decided to seek the nearest port where repairs could be made. Accordingly we went to Las Palmas, in the Canary Islands, and there had the dock strengthened so that it was more seaworthy than when we left home with it. But for this work I fear we would have lost the dock, as soon afterward we had a gale in the Mediterranean sea worse than any we saw in the Atlantic, and which, I believe, would have taken the end pontoons off the dock had they been allowed to remain in their crippled condition.

When we arrived at Las Palmas we were just fifty-seven days from the Patuxent River, and had dragged the dock in that time 3,844 miles, or considerably more than the distance across the Atlantic could we have applied it to a direct course. Had it not been necessary to repair the dock we should have been obliged to stop at the Canaries anyhow, because our long period at sea had almost exhausted the supply of coal and water on the dock. We started with what was considered ample to reach Port Said, but the many breakdowns had caused the use of the capstans and other steam machinery on the dock much more than had been expected, and, as no condenser had been provided, a great amount of fresh water had been lost in the form of exhaust steam. There was an evaporator on the dock, but without a condenser for the steam that passed through it it is doubtful if it produced as much water as was required in the form of steam to evaporate the sea water in the first place. Some scholarly member of this Society familiar with thermal units and their habits may be able to cipher this out otherwise, but to an ordinary mind this peculiar form of an evaporator looks remarkably like a

device for making something out of nothing. At any rate, its use was discontinued when it became evident that it caused a loss and not a gain in the fresh-water tanks.

Having referred to people on board the dock, I will answer a question that has been asked me a number of times already, and say that there always were men on the dock when it was being towed at sea—more than thirty of them, in fact. There were nine men, employees of the Bureau of Construction and Repair, who constituted a permanent crew for the dock to attend to its care and operation when it reached its destination. Then there was a regular crew shipped for the voyage the same as the crew of any merchant ship, composed of sailing master, two mates, cook, steward, twelve seamen (afterward increased to eighteen), a rigger and a few others, such as messmen and telegraph operator. There was nothing perilous in their position; on the contrary, they were undoubtedly safer than they would have been on a real ship, and they were certainly more comfortable in rough weather than any of us were on the towing ships.

April 7 to 9, when in the eastern basin of the Mediterranean Sea about 200 miles east of Malta, we encountered a gale from eastward and southward that at one time reached the force of a whole gale (10 by the Beaufort scale) and raised a heavier sea than any that we had experienced in the Atlantic Ocean. The *Glacier*, as she had done on several previous occasions, cast off to reduce the strain on the tow lines, but the gale steadily grew in force, and the second day, April 8, the dock broke adrift by the parting of a large shackle that joined the legs of the towing bridle to the first span. The dock drifted rapidly to leeward broadside to in the trough of the sea and achieved her maximum roll on this occasion, only eight degrees, but owing to the great height of the side walls above the water this looked very much more to an external observer. As a lee shore was only 200 miles away the situation was more critical than when the dock was adrift in mid ocean, but after it had been loose twenty-three hours and while the sea was yet heavy the *Brutus*, aided by the *Potomac*, got a

line to it and was able to check drifting to leeward though unable to tow it against the weather. By the 10th the storm had blown itself out and we all resumed towing to the eastward. The evening of the 11th, when towing in a smooth sea with no wind, a double 15-inch manila span between the *Brutus* and the dock suddenly parted, from no cause whatever except what is popularly known as "pure cussedness," and set the dock adrift again. The situation offered no difficulties in picking up the dock, but the work of hauling in the heavy lines, ranging them clear for running, and running them again in making up the tow, kept us busy nearly all night, and it was after two o'clock in the morning before we were steaming ahead on the course again.

The passage of the dock through the Suez Canal was effected with less trouble than anticipated, and there would have been practically none had not an unusually strong beam wind forced it out of one of the sidings where it was moored and against the bank on the opposite side, blocking the canal for several hours. It was towed through by two powerful tugs of the canal company in tandem ahead, with the *Potomac* astern to push and to prevent the rear end from yawing into the banks. The dock was in the canal four and one-half days, but was actually under way only thirty-five hours, considerable periods having been spent at anchor in Lake Timseh at Ismailia and in the Great Bitter Lake to allow steamers to pass both ways and avoid stopping traffic. The *Potomac* rendered such excellent service here that it made up for all the trouble she had caused us in the Atlantic. She returned to the United States from Suez.

At Port Said we got the 8-inch steel hawsers that had been ordered from England, and the *Brutus* and *Caesar* put them on their towing drums in place of the 6-inch wires. In making up the tow at Suez the manila spans between the *Brutus* and the dock were discarded and their place filled by 105 fathoms of 2½-inch chain taken from the spare bridles on the dock. One end of this chain was shackled to the dock's bridle and the other into the end of the 8-inch wire from the

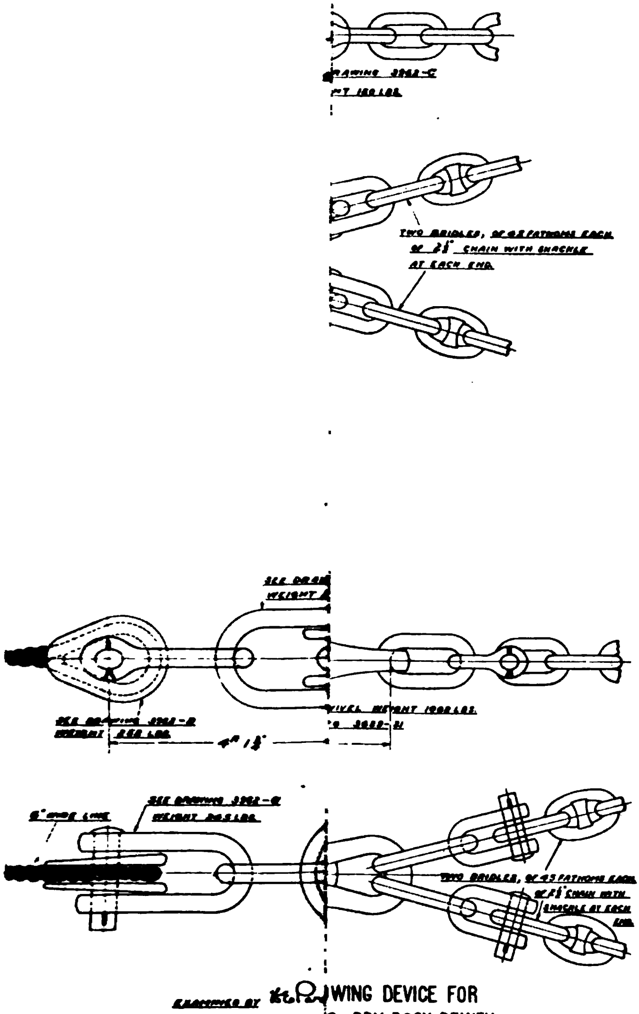
Brutus' towing drum. I neglected to state in the proper place that at Las Palmas we received from the United States spare drum gears for the towing engines, and those machines on the *Brutus* and *Caesar* were put in good order again. The weight of the chain span just described was fifteen tons and of the 8-inch wire joined to it almost five tons, the two forming a catenary so deep that there was no danger of any power we had ever bringing it up taut. The *Caesar* had a double manila span between her 8-inch wire and the *Brutus*, and the *Glacier* used another one to the *Caesar* from a new 6-inch wire on her towing machine. With the towing train thus strengthened we felt free to steam at full power, and did so with such excellent results that we towed the dock from Suez to Singapore, 5,000 miles, in forty-eight days, without a break or accident of any sort to the towing gear, and with only one stop, of two hours' duration, to replace a gasket blown out on the *Glacier*.

In entering the port at Singapore the combination of a very dark night and a swiftly-running spring tide threw the squadron into some confusion while shortening in the lines, and when the ships got straightened out clear of danger from each other the dock was found to be adrift. Contrary to all the rules of dynamics, the big double shackle joining the chain span to the dock's towing bridle had parted, but how it happened was not at all clear. The dock was anchored, and after daylight was gotten under way and towed into port with no greater loss than a few hours' time, but as it had taken us six months to get that far, time had become a matter of small value to us. A stop of a week was made at Singapore to coal and water the ships and to give the crews a run ashore. Then we resumed our voyage, proceeding northeastward into the China Sea, where the best long run of the whole trip was made, the distance of over 1,300 miles to Olongapo being made at the average rate of 112½ miles per day. The best day's run of the whole voyage was 152 miles, made June 9th in the Bay of Bengal, where a current of at least forty miles per day favored us; the worst day's run was, as before mentioned, in the Atlantic,

where we achieved *minus* 24 miles one day steaming against a head wind and sea. We arrived at Olongapo in Subig Bay early in the morning of July 10, 1906; the towing ships let go one after the other, the dock was anchored, and a few hours later we left it there without regret and returned to the world that we had been out of for more than half a year.

Passages.	Miles made good.	Time.		Daily averages.
		Days.	*Hours.*	
Patuxent River to Las Palmas,	3,844	56	23	67.56
Las Palmas to Port Said, .	2,849	32	00	89.03
Port Said to Suez, . . .	87	1	11	58.78
Suez to Singapore, . . .	4,992	48	7	103.38
Singapore to Olongapo, . .	1,317	11	17	112.47
Totals, ...	13,089	150	9	87.03

Ship.	Coal consumed.	Cost of coal.
	Tons.	
Glacier, ...	5,136	$27,224
Brutus, ...	3,664	12,594
Caesar, ...	3,238	14,164
Dewey, ...	923	4,088
Totals, ...	12,961	$58,070



EXAMINED BY

WING DEVICE FOR

S. DRY DOCK DEWEY

NT DEPT., U. S. NAVY YARD BOSTON.

NOV. 21, 1906.

DRAWING NO 3971-36.

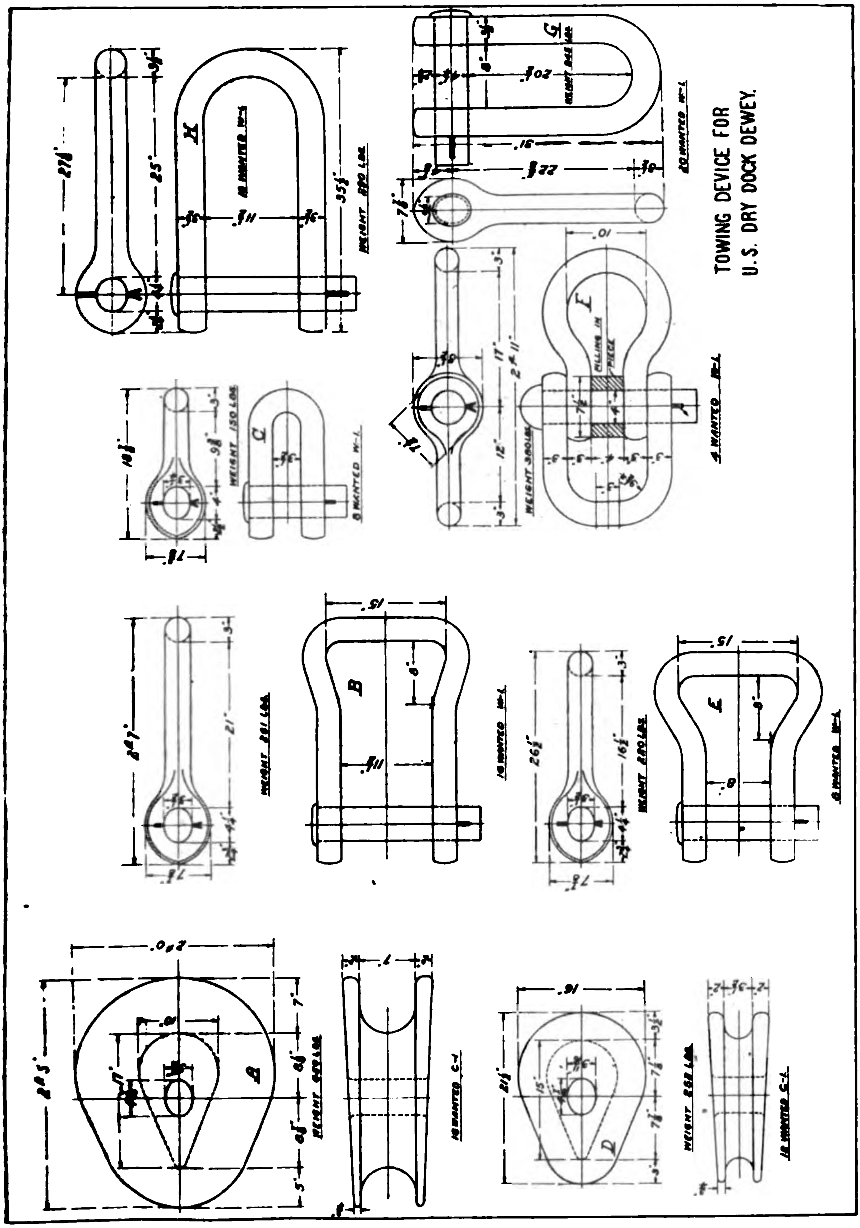


Plate II.—Details.

CORROSION OF STEEL BOILER TUBES ON VESSELS FITTED WITH TURBINE ENGINES.

By Commander J. Edward Palmer, U. S. Navy, Retired, Member.

A very curious and interesting case of unusual corrosion of steel boiler tubes on a vessel fitted with turbine engines has recently been brought to light, and the writer is indebted to the officers of the Shelby Steel Tube Company, of Pittsburg, Pa., through the courtesy of whom this information was obtained.

The facts are as follows:

The steam yacht *Tarantula* was first equipped with Parsons turbine engines and Yarrow boilers. As these boilers did not furnish sufficient steam they were replaced in the Winter of 1904 and 1905 by Mosher boilers, furnishing more steam and at a somewhat higher pressure of 275 to 300 pounds per square inch. These boilers were tubed with Shelby seamless steel tubes of No. 14 and No. 16 gauge, the first five rows nearer the fire being of the heavier gauge.

After the first two months' service complaint was made by Mr. Charles Mosher that tubes were pitting badly and several had given out. The Shelby Steel Tube Company immediately instructed their metallurgical engineer to examine closely into this matter, and if possible determine the cause of this unusual corrosion, which resulted in the following interesting information.

By the end of the second season (which means after a total service of about four months) four tubes had given out in the forward boiler and twenty-one in the after boiler, all of these being of the thinner (No. 16) gauge.

The owner having decided to retube, an opportunity was

given to examine the interior of a number of the tubes which had not failed. The samples when opened up from end to end showed quite an irregular effect, some being practically good, others pitted at certain places, mostly near the ends, and others again more or less pitted throughout. A composite sample of the unpitted and pitted tubes in each boiler was carefully taken and gave the following results on analysis, showing practically the same composition of steel.

FORWARD BOILER.

	Sulphur.	Phosphorus.	Manganese.	Carbon.
Good tube, .	.035	.008	.41	.14
Pitted tube, .	.026	.007	.40	.15

AFTER BOILER.

	Sulphur.	Phosphorus.	Manganese.	Carbon.
Good tube, .	.031	.007	.50	.15
Pitted tube, .	.031	.005	.41	.16

It was very noticeable that a brownish deposit had collected in the pitted portions of the tubes, forming little nodules over the pits. This deposit was analyzed, and invariably showed copper when taken from the pitted portions, while the deposit in the unpitted tubes gave no sign of this foreign metal.

The following average analysis of these boiler residues is interesting:

	From after boiler.	From forward boiler.
Iron, ...	51.20	58.06
Copper, ...	1.84	.39
Zinc, ...	None	None

The effect of this copper, even in comparatively good water, would tend to start a local galvanic current between the iron and the copper where they were in contact, thereby slowly eating away the tubes in pits.

As it was evident that this corrosion was caused by the above, it remained to be discovered how this copper came to be accumulated in the boilers. After following carefully up every clue all the evidence pointed to the blades of the tur-

bine engines, which were made of a bronze containing a considerable percentage of copper. These blades were found to be eroded by the steam impinging on them at a very high velocity, and it is most probable that the copper was carried over by the steam (possibly with the aid of volatile acids) into the condensers, and from thence into the boilers and accumulating in the tubes.

The deposit found between the turbine blades was found to carry from 1 to 2½ per cent. of copper, while that in the mud drum of the after boiler at the end of the Summer of 1905 showed 3.6 per cent. copper.

The boiler scales showed marked evidence of the water or other boiler additions having carried organic matter, the decomposition of which may have had much to do with increasing the action of the steam on the bronze blades of the turbine.

Tubes of exactly the same composition and manufacture as the above, and similar to them in all respects, have been repeatedly used on vessels fitted with reciprocating engines without showing any such pitting, as for an example, on the yacht *Arrow* these tubes gave excellent service for about eight years without any signs of such corrosion as was found on the *Tarantula* after a few months' service.

From all the facts and information that could be obtained in this case there seems to be no doubt whatever that the copper which caused the pitting on the tubes was taken by the steam from the turbine blades and carried over into the boilers through the condensers and deposited on the tubes.

While of course this is very serious, and, should it have occurred on a new large ship, would have been most costly, yet it seems to be a difficulty which could easily be overcome by some experimenting. It has already been suggested to make the turbine blades of a material the metal of which, if deposited on the boiler tubes, would not have an injurious effect, or to place iron or steel turnings in the hot well, or at some point between the turbines and the boiler, which would take up any deposit of injurious metal before it reached the tubes.

TRIALS OF THE *LUBECK*.

BY R. VEITH, CONSULTING DIRECTOR IN NAVAL ARCHITECTURE AND MARINE ENGINEERING, VULCAN WORKS, STETTIN.

(Translated from "Marine Rundschau" by Commander R. S. GRIFFIN, U. S. Navy, Member.)

On account of the extended use of turbines for driving dynamos, the German Government entered into contract with the Vulcan Company for the construction of the *Lubeck*, which was to have turbines furnished by the German Parsons Marine Turbine Co. That the results of her trials are only now available is due to the fact that the trials covered much time, involved considerable work and required great elaboration and the sifting of many data.

The contract required that the ship, in a 6-hour trial under forced draft, not exceeding 2½ inches of water, should make a speed not inferior to that of the sister ship *Hamburg* of 10,000 I.H.P., fitted with reciprocating engines and the same boiler installation, and built also by the Vulcan Company; that the coal consumption on the high-power endurance trial and on the coal-consumption trial should not be greater than 1.98 pounds per I.H.P., the I.H.P. to be taken as that developed by the *Hamburg* at the same speed; that on the high-power endurance and the consumption trials the turbines should be run at such a speed as would correspond respectively with about 7,000 and 1,400 I.H.P. for the *Hamburg;* and that the draught of water and other conditions should be as nearly as possible the same for both ships.

The principal dimensions are:

Length between perpendiculars, feet,	340.56
Breadth, extreme, feet,	43.31
Draught, mean, feet,	16.40
Displacement, metric tons,	3,217.2

The boiler installation consists of ten bent-tube water-tube boilers, without superheaters, and of the type generally used in the German Navy (Schultz-Thornycroft.—Ed.), having a collective grate and heating surface of 580 and 29,563 square feet, respectively. The steam pressure is 213 pounds.

The turbines are in two compartments separated by a middle-line bulkhead, and consist of two sets of main turbines and two cruising turbines for ahead motion, and of four backing turbines for astern motion. The arrangement is shown on the accompanying sketch.

Of the main turbine sets, one is in each of the starboard and port engine rooms. Each set consists of a H.P. and a L.P. turbine, each of which operates a screw shaft, thus making four shafts in the ship. Each set is entirely independent of the other, and is used for running at high power. For low and moderate power, the main are coupled up with the cruising turbines, the latter of which consist of a H.P. and a L.P. turbine, the H.P. located in the starboard and the L.P. in the port engine room. They operate the inboard shafts on each side, being on the same shaft as the main L.P. turbines.

The four backing turbines are located one on each shaft, the two inner ones being in the casing of the L.P. main turbines. Originally, the backing turbines were so arranged that both turbines on each side received steam at the same time and had to be operated together. During the trials, however, a stop valve was, for economical reasons, placed in the steam pipe to the inner backing turbine, so that the outer ones could be operated alone.

Originally, each of the four shafts was provided with two three-bladed screws of the following dimensions:

Diameter, feet, ...	4.5
Pitch, feet, ...	4.5
Surface, projected, square feet, ...	6.375

In addition to these, the Parsons Company provided, for experimental purposes, four other larger screws of the following dimensions:

Diameter, feet, ...	5.578
Pitch, feet, ...	4.918
Surface, projected, square feet, ...	9.497

All screws were of manganese-bronze.

Three series of tests were made with these two sets of screws:

1st. Trials with eight small ones,

2d. Trials with four large ones,

3d. Trials with four large and four small ones.

Besides the auxiliary machinery, which is the same as in the *Hamburg*, there are in each engine room two lubricating oil pumps, with cooling and filtering connections, for forcing oil through the turbine bearings.

The air pumps are independent Weir "wet" pumps, while in the *Hamburg* they are worked from the engine. Dry-air pumps are not fitted, as in torpedo boat *S-125*. (See page 961, JOURNAL, Vol. XVIII.)

To obtain economy at low and moderate speeds the turbines may be coupled in three ways:

For low power—up to about 14 knots—the H.P. and L.P. cruising turbines are coupled up with the H.P. main turbines, from which latter the steam goes to the condenser by way of the L.P. main turbine. (1st Combination.)

For moderate power—up to about 18.5 knots—only the L.P. cruising turbine is coupled with the H.P. main turbines. (2d Combination.)

For maximum power—both cruising turbines are disconnected and steam is admitted direct to each H.P. main turbine. (3d Combination.)

Reduction of power in each of the arrangements is obtained by throttling. Increased power may be obtained by admitting live steam to the L.P. cruising turbines in the 1st Combination and to the H.P. main turbines in the 2d Combination.

Föttinger torsion meters, which had previously been tested, were mounted on each shaft, in order to measure the power developed.

The coal used was weighed before being put in the bunkers,

and upon completion of the trial that remaining in the bunkers was again weighed in order to get the amount actually used. A check was also kept on the expenditure by means of the usual bucket method.

TRIALS.

I. CONTRACT TRIALS.

The contract trials were made with the eight small screws.

(*a*) *Six-hour forced-draft trial.*—The requirements were as noted on page 57, and as the measured-mile trials of the *Hamburg* gave 21.9 knots at 10,000 I.H.P., this was fixed as the contract speed for the *Lubeck.* The mean revolutions for this trial were 662.5, which were equivalent to a speed on the Eckernförde mile of 22.25 knots, or .35 knot more than required.

The turbines and boilers worked without derangement.

(*b*) *Twenty-four-hour coal-consumption trial at cruising speed.*—The measured-mile trial of the *Hamburg* gave 12.4 knots with 1,400 I.H.P., and to reach this speed it was necessary for the *Lubeck* to run at 333.5 revolutions per minute. The coal consumption was 2.204 pounds per I.H.P., or 0.22 more than the contract permitted.

The trial was made with the turbines coupled as in the 1st Combination.

The machinery gave no trouble whatever during this trial.

(*c*) *Twenty-four-hour high-power endurance trial.*—On her endurance trial the *Hamburg* made 20.2 knots with 7,000 I.H.P., and this was the speed fixed for the *Lubeck*, the coal consumption to be the same as that stipulated for the preceding trial. This trial was made with the turbines arranged as in the 2d Combination, with additional steam in both H.P. main turbines. The revolutions required for the speed were 568 per minute, and the average made for the twenty-four hours was 567.3. The coal consumption was slightly in excess of the limit, being 2.13 instead of 1.98 pounds per I.H.P.

(*d*) *Twenty-four hour high-power endurance trial repeated.*—The contractors took exception to the preceding trial, because

a portion of it was run in much shallower water than that on the Eckernförde mile, and, therefore, required a marked increase in power in order to obtain the required number of revolutions. So, the trial was repeated under the following conditions:

"The trial must not be run if the force of the wind exceeds 3. (On coal-consumption trials, the usual limit is 4.)

"Before commencing the trial the fires must be cleaned, and the ship brought to such a draught that the mean draught for the trial shall be as nearly as possible the same as that of the *Hamburg* on her measured-mile trials.

"The mean revolutions will remain as before, 568.

"Should the force of the wind reach 4, the remainder of the trial will not be based on revolutions, but will be run with the same initial pressure in the H.P. main turbines as had been maintained from the commencement of the trial to the time that bad weather set in."

The trial was made on the 8th and 9th of December, 1905, with the stop valve to the L.P. cruising turbine wide open (2d Combination) and some live steam admitted to the H.P. main turbines. For ten hours the force of the wind was 3 to 4 and at times 4 to 5, and during this time the initial pressure in the H.P. main turbines controlled the trial condition.

Turbines and boilers worked satisfactorily. The mean revolutions per minute were 566.5, and the coal consumption was 1.92 pounds, .06 pounds less than the requirement.

Less steam was admitted to the H.P. main turbines than on the former trial, the initial pressure being about 65.4 as against 72.5 pounds. The boiler pressure and the initial pressure in the L.P. cruising turbine were the same on both trials.

The coal per I.H.P. on the trials of the *Hamburg* was 1.91 pounds on the endurance trial at cruising speed, and 1.77 on that at full speed.

ADDITIONAL TRIALS.

Besides the contract trials, a large number of others was made to determine the influence of different screws and turbine combinations on the speed, coal consumption and maneuvering qualities of the ship.

The trials were divided into four groups, depending upon the propellers and their arrangement, and comprised measured-mile and coal-consumption trials, and determination of the distance the ship will go from full speed ahead to stop under the influence of the backing turbines.

Group I embraces all trials with the original arrangement of propellers—two small ones on each shaft. The highest speed was 22.369 knots, with 672 revolutions and 13,705 shaft horsepower; the slip was 25.11 per cent.

Group II embraces all trials made with the four large screws, which were made with one on each shaft mounted on the after cone. The maximum speed was 22.39 knots with 623 revolutions, 13,029 shaft horsepower, and a slip of 25.97 per cent.

Trials on the mile at Neukrug in deep water, were also made with these screws. The maximum speed there was 23.16 knots with 14,158 shaft horsepower and a slip of 26.5 per cent.

Group III embraces those trials made with two screws on each shaft—a small one forward and a large one aft, and, therefore, with four small and four large screws. The large screws on the outer shafts had to be reduced in diameter about 3.937 inches in order to clear the small screws on the inner shafts in the same athwartship plane. With this arrangement of screws was obtained the greatest speed in relation to power. The speed was 22.562 knots, with 601 revolutions and 13,573 shaft horsepower. The slip was 15.51 per cent. for the forward and 22.67 for the after screws.

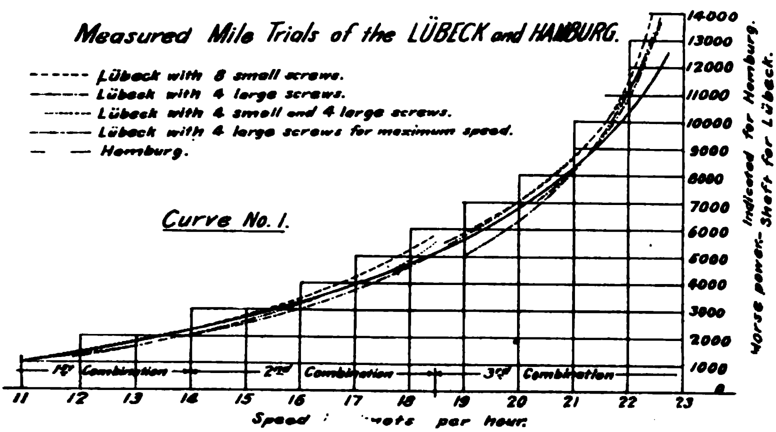
Group IV embraces those trials made with one of the three sets of screws furnished by the Parsons Company. They were built by this company and were expected to give a speed in excess of what had hitherto been obtained, and were of the following dimensions:

	For outside shaft.	For inside shaft.
Diameter, feet, ...	5.25	5.75
Pitch, feet, ...	4.70	5.17
Surface, projected, square feet, .	12.95	15.53

The screws were placed as in Group II, on the after cone of each shaft.

The maximum speed was 22.551 knots, with 625 revolutions and 13,879 shaft horsepower. The mean slip was 25.79 per cent.

The results of the trials over the Eckernförde mile are given on the accompanying Curve 1.



Trials with the large screws on the forward cone of the shaft were out of the question on account of the great cost of alteration in shafts and ship's hull. Measured-mile trials in deep water with propellers other than those of Group II were not made, because it was to be expected that no great difference in speed would be obtained.

The coal-consumption trials were in general of six hours' duration. Before each trial the boilers were thoroughly cleaned, so that it would not be necessary to clean fires during the trial, and the evaporators were not in use. With the eight small screws several trials were made at different speeds and with different turbine combinations. In order to obtain information on the effect of the 3d Combination on coal-consumption some trials were also made with steam to the turbines highly throttled.

Several coal-consumption trials were also made with the four large propellers of Group II.

In the accompanying Curve 2 are given the hourly consumption of coal during the contract coal-consumption trials of the *Lubeck* and *Hamburg*, alongside that of the *Lubeck* on her forced-draft trial and on the unofficial coal-consumption trials.

Special mention should also be made of two coal-consumption trials made on the 25th and 26th and on the 28th and 29th of September, 1905, with the *Lubeck* and *Hamburg* steaming in company. In order to equalize the effect of wind, sea and current, each ship led for six hours, the other following in her wake. Both were brought to the same draught at the beginning, and the *Lubeck* was run with the four large screws of Group II, each 5.578 feet diameter and 4.918 feet pitch.

During the first trial the speed was about 20 knots. *Lubeck* ran under the 2d Combination and with live steam in both H.P. main turbines, and the steam to the L.P. cruising turbine wide open.

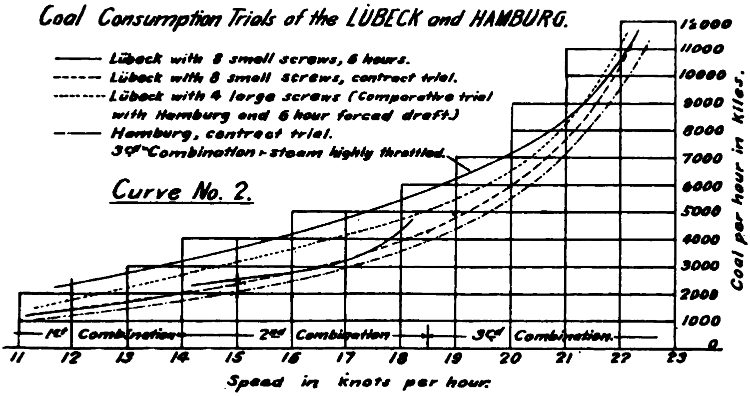
	Lubeck.	*Hamburg.*
Revolutions per minute, ...	508	—
Horsepower, indicated, ...	—	7,027
shaft, ...	7,662	—
Coal per I.H.P., pounds, ...	2.09	2.14

Force of wind, 3 to 7; state of sea, 3 to 5.

The high consumption of the *Hamburg* compared with that of the *Lubeck* is explained by the fact that the firemen of the former had had very little experience in firing on coal-consumption trials.

The second coal-consumption trial was made at a speed of about 15½ knots. The *Lubeck* operated under the 2d Combination, and developed 3,297 shaft horsepower with 390 revolutions. The initial steam pressure in the L.P. cruising turbine was throttled to about 121 pounds. The I.H.P. of the *Hamburg* was 3,303. The wind was 3 to 5 at the beginning, but died out entirely during the trial, which ended after twenty-three hours on account of a dense fog.

The coal consumption of the *Lubeck*, based on the I.H.P. of the *Hamburg*, was 2.25 pounds, and that of the *Hamburg*, 1.70. The results of these trials of the *Lubeck* and of a six-hour trial under forced draft, with the same screws, are given in Curve 2.



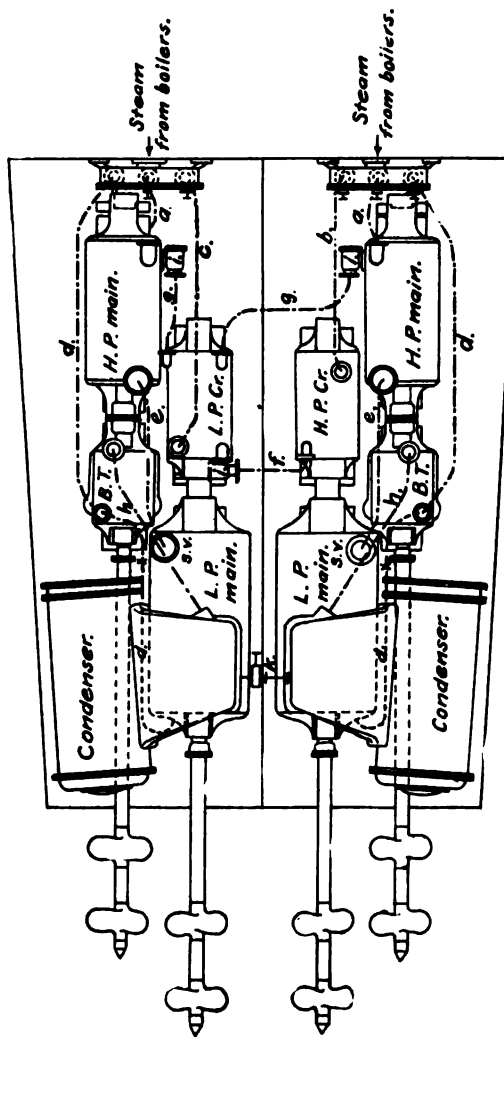
In order to show the influence of wind and sea on coal consumption at the same speed, the results of the last eight hours of the trial, during which there was practically a calm, and of the first fifteen, when the wind was 3 to 5, are shown separately. The result was a consumption for the *Lubeck* of 2.0 pounds per I.H.P. during the calm and of 2.47 pounds when the wind was 3 to 5. For the *Hamburg*, no noteworthy difference was apparent.

TRIALS TO STOP HEADWAY.

These were made with each of the four arrangements of propellers, and the results are given in the accompanying table. On account of the large steam consumption of the backing turbines, and to prevent too much drop in steam pressure, the opening of the steam valve to the turbines was so regulated that the drop should not exceed 43 pounds.

The figures in the table are for the simultaneous working of all four backing turbines. Those for the working of the two outer backing turbines only were also determined. The

Steam connections of the Turbine Installation of the LÜBECK.



a-a. Steam to H.P. main turbine.
b. Steam to H.P. cruising turbine.
c. Steam to L.P. cruising turbine.
d-d. Steam to Backing turbines.
e-e. Exhaust from H.P. main to L.P. main turbines.
f. Exhaust from H.P. to L.P. cruising turbines.
g-g. Exhaust from L.P. cruising to H.P. main turbines.
h-h. Exhaust from Backing turbine to condenser.
i-i. Non-return valve.
K. Connection between both L.P. turbines.
S.V. Stop valve in steam to backing turbines on inner shafts.

difference in time from full speed ahead to dead stop of the ship when backing full speed with the two outer and with the four backing turbines was 23 seconds, and the distance run about 328 feet more. Naturally, the time of stopping cannot be accurately determined on account of the difficulty of telling when the ship is absolutely at rest.

Number of boilers.	Speed of ship at the time signal "Full speed astern" was given.	Distance run in feet before ship was dead in water.				
		Lubeck.				*Hamburg.*
		Eight small screws.	Four large screws.	Four large and four small screws.	Four large screws (for maximum speed).	
10	Slow speed, 5 knots	335	171	164	246	184
10	Half speed, 9 knots	384	413	361	480	361
10	Moderate speed, 11 knots	755	692	636	700	590
10	Maximum speed, 22 knots	1,430	1,765	1,529	1,640	919

CONCLUSIONS.

The *Hamburg* showed a speed of 22.535 knots on the Eckernförde mile and 23.15 in deep water on the Neukrug mile. The maximum speed of the *Lubeck* is, therefore, practically the same as that of the *Hamburg*. The power developed by the engines of the two vessels are, however, quite different, the turbines of the *Lubeck* having shown in shallow water 13,573 shaft horsepower in Eckernförde Bay and, in deep water, 14,159 shaft horsepower at Neukrug, while the engines of the *Hamburg* under similar conditions developed respectively 11,889 and 11,582 I.H.P. This indicates that the turbine ship required greater power than the one with reciprocating engines. That the *Lubeck* did not get greater speed with this increased power is attributed to the inefficient working of her screws.

The coal consumptiom of the *Lubeck* was higher than that of the *Hamburg*, as will be seen by the following:

Coal used by both ships referred to the I.H.P. of *Hamburg*.	Cruising speed.	Maximum endurance speed.
Lubeck, ...	2.20	1.92
Hamburg, ...	1.91	1.76

During the forced-draft trial the coal per hour of the *Lubeck* was 25,117 pounds at 22.265 knots, and that of the *Hamburg* 22,185 at 22.2 knots. The coal for auxiliary purposes is not included. While these figures show a radius of action for the *Hamburg* superior to that of the *Lubeck*, it is to be expected that, with continued use, the valves and pistons of the *Hamburg* will show more leakage, but that the radius of action of the *Lubeck* will, nevertheless, be inferior to that of the *Hamburg*. Thus far, in squadron, the coal consumption of the *Lubeck* has been about 8 per cent. greater.

The distance the *Lubeck* goes from full speed ahead until dead in the water under the influence of her backing turbines is much greater than for the *Hamburg*.

MISHAPS DURING THE TRIALS.

During the long period covered by the trials the *Lubeck* had only two accidents, and each case was one of serious blade injury. The first time, the rotor and stator blades of the H.P. cruising turbine were broken from the 11th to the 18th row; the second time, the second-stage blades of the L.P. cruising turbines were ground and partly bent. In each case the repairs required about fourteen days.

The clearance between the blades and the casing was then increased, and no more accidents occurred; this increased clearance apparently had no effect upon the economy of the system.

Upon completion of the trials the turbines were examined, and, except for injuries to blades occasioned by wire nails left in the casing after casting, only a few blades were slightly

ground off. The edges of the steam gland collars in the casing of the backing turbines showed knife edges and were partly ground off; one was broken. Those on the shaft of the backing and L.P. main turbines were also ground off.

WEIGHT AND SPACE REQUIRED.

The combined weight of the turbine and boiler installation of the *Lubeck*, exclusive of the auxiliary machinery, is 609 tons, and the corresponding weight on the *Hamburg*, 652.7 tons. The weight in the *Lubeck* is, therefore, 43.7 tons, or 7 per cent. less than in the *Hamburg*. Disregarding the boilers, which are the same in each ship, the weight of the engine installation of the *Lubeck*, including the necessary auxiliary machinery and appurtenances, is 271 tons, while that of the *Hamburg* is 323 tons, or a reduction in weight of 16 per cent. for the *Lubeck*. The turbine also tends to economy in another direction, in that the protective deck can be worked lower than with a reciprocating engine.

This economy in weight of turbine engines will not be obtained in the turbine ships now under construction, on account of improvements that have been found to be necessary.

The turbine engines of the *Lubeck* are placed in the same athwartship space as the reciprocating engines of the *Hamburg*, but the entire machinery of the *Lubeck* could not, however, be so favorably placed, and it was, therefore, necessary to place the transverse bulkhead two frame spaces further aft than in the *Hamburg*, thus entailing undesirably large spaces.

FURTHER CONCLUSIONS.

As there are no heavy reciprocating parts on the *Lubeck* as on the sister ship, so there is an absence of the vibration noticeable in the latter. This secures a steady gun platform and a steady compass, by which means improved aiming and steering are obtained. The low position of the turbines and screws secures better protection for them.

So far overhauling has been less necessary than with reciprocating engines, so it is fair to expect a greater readiness for service of ships with turbines.

Turbines are easily maneuvered. The operation is simpler than with reciprocating engines because it is merely the opening and closing of a valve, which does away with a reversing gear. There is an absence of annoyance caused by water in the cylinders when maneuvering, because the water is broken up into very small particles.

The small attention necessary during the working of the turbines, the little overhauling and cleaning while idle, which advantages (on account of the steam being free from oil) extend to the boilers, condensers, feed-water heaters, etc., save the personnel. Injury to attendants is unknown with turbines.

On account of the high temperature and excessive moisture in the engine rooms of the *Lubeck*, improvement in these conditions may be looked for. A reduction in personnel was not attempted. Whether improvements in turbines will bring about such a reduction must be left to further experience.

CONCLUSION.

The trials of the *Lubeck* have furnished most valuable information, but they have not been entirely conclusive, because the ship had to take her place in squadron. She is the first of our larger ships to have turbines. If, in competition with the highly developed reciprocating engine, it has shown a few defects, this beginning may, taken all in all, be considered a success.

As an evidence of its preference for turbines, and for further investigation of the turbine question, the German Admiralty has recently contracted for turbines for the small cruisers *Ersatz Wacht* and *Ersatz Komet*, notwithstanding the fact that they had to pay a much higher price for turbines than for reciprocating engines. The subsequent cost of maintenance will more than balance this first cost. Parsons turbines have

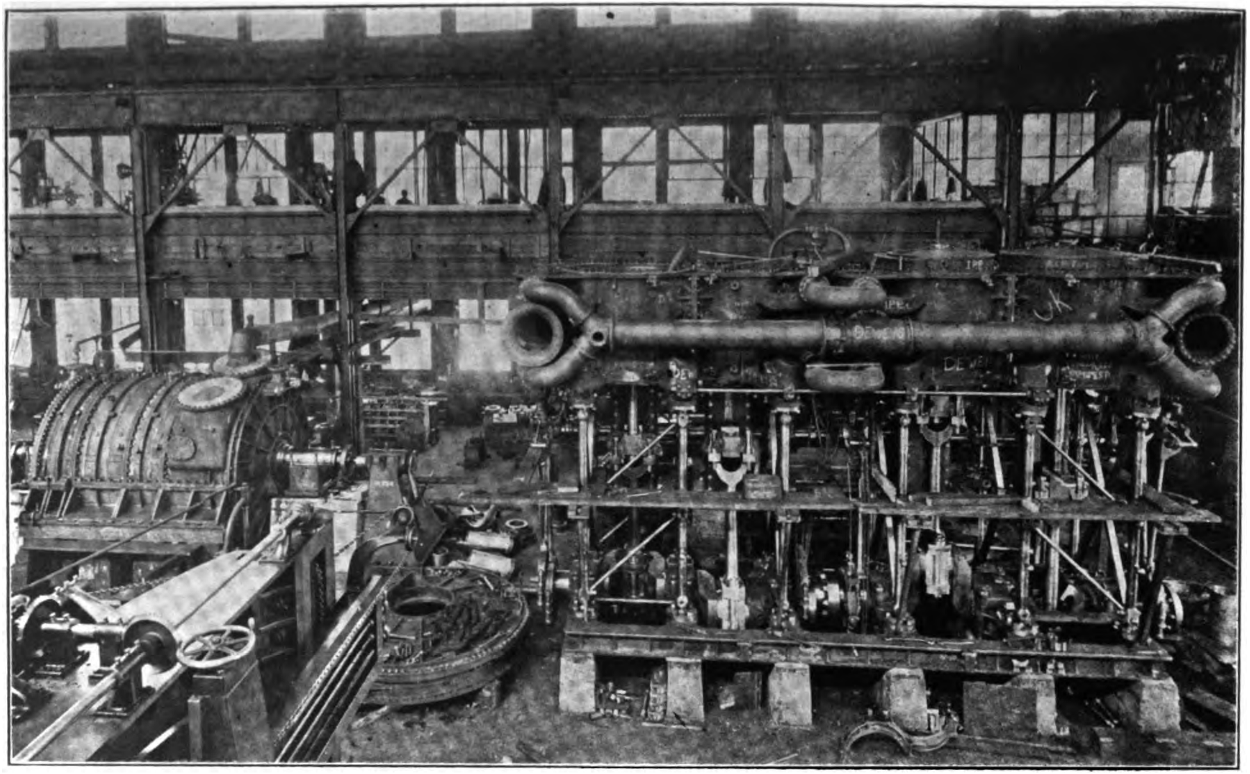
been decided upon for both ships, because other turbines for large ship installations are still in the experimental stage. Whether the Parsons will be the only turbine used in our Navy remains to be seen; others may, in short time, reach such a stage of development as to be able to compete with it.

CHARACTERISTICS IN DESIGN AND ARRANGEMENT OF MARINE TURBINES AND PROPELLERS.

By Ernest N. Janson, Member.

II.—CURTIS SYSTEM.

Apart from the turbines and their action, machinery installations with Curtis steam turbines for propelling purposes differ from Parsons principally in the division of the total power transmitted to the propellers. As now constructed turbine arrangements of Parsons type consist generally of three units, often of four, but rarely of two, this latter arrangement having been used in only one instance, so far as known, namely, in that of the *Narcissus*, a steam yacht of small power. With Curtis type the prevailing practice comprises the regular twin-screw system, with two turbines and two shafts. Where the power needed is small, as in steam launches, a single shaft driven by one turbine is most suitable. Arrangements of three or four shafts with an equal number of turbines have not as yet been proposed, the reason of which may be attributed to certain inherent features in this type of turbine, rendering a twin arrangement more desirable. Enhanced cost and excess in weight as well as space over the present system would perhaps also follow. Each turbine being entirely self-contained, or, in other words, having within a single casing the parts corresponding to the high-pressure and low-pressure turbines as well as reversing turbines of Parsons' system, would, with a multiple shaft arrangement, properly require independent condensers, air pumps and circulating pumps for each unit, and, therefore, considerably add to the number of auxiliaries. In very large installations, however, such as in the *Mauretania* and *Lusitania*, where the auxiliary units, on



CURTIS MARINE TURBINE U. S. S. "SALEM."
8,000 B.H.P.

4-CYL. TRIPLE-EX. MARINE ENGINE U. S. S. "VERMONT."
8,250 I.H.P.

account of the extraordinary power, in any event have to be divided up, it is not probable any great difference between the two systems as to auxiliaries would appear, other things being equal.

Principal exterior dimensions of the turbine are primarily influenced by the speed of revolutions partly determined by the propeller, therefore indirectly by horsepower and speed of ship. Initial steam pressure and the pressure at the condenser do, in a measure, determine the length, but are of greater moment with respect to internal dimensions.

Illustrations in half tone facing page 72 show comparatively the space occupied by an engine and a Curtis turbine. The following dimensions will explain:

TURBINE.

120 inches, 7-stage, 350 R.P.M.

Length over all, casing	16 feet 2¾ inches.
Width over all, casing	13 feet 6 inches.
Height over all	12 feet 6 inches.
C to C, main bearings	18 feet 6 inches.
Weight	102 tons.

ENGINE.

32½ inches × 53 inches × 61 inches × 61 inches × 48-inch stroke. 120 R.P.M.

Length over all, cylinders	33 feet 6½ inches.
Width over all, cylinders	11 feet 3 inches.
Height	21 feet 9 inches.
C to C, end bearings	25 feet 3½ inches.
Weight	153 tons.

The usual pressure employed at the turbine inlet is about 265 pounds absolute, but may be almost any reasonable quantity within practicable limits of boiler and steam-pipe constructions. It must be remembered that with this type of turbine the pressures are confined initially within small nozzles and that the casing receives the steam after it has been expanded. The vacuum should be as high as possible, 28 to 29 inches being aimed at, and the former, at least, is readily obtained. In this connection it is interesting to point out that a vacuum as high as 28 inches is obtained in marine turbine arrange-

6

ments with an air-pump capacity less than that required for reciprocating engines running with 27 inches, and this without additional apparatus either in form of "augmentors" or dry-vacuum pumps. The reason is plainly to be found in the steam-packed glands in the turbine, therefore fewer air leaks. Condensers of large surface, together with a considerable amount of cooling water, are invariably required to adequately condense the steam. As an instance may be cited a large ocean-going turbine steamer now running equipped with both dry and wet-air pumps, maintaining a steady vacuum of 28½ inches with the latter alone in operation, the dry pump having no function except its presence. The condensers, however, are large, having about 1.6 square feet of cooling surface per equivalent I.H.P.

The importance of simultaneously considering the design of propeller and turbine having been discussed in a previous article, it is now necessary only to transform the data there given to conform with the new condition, namely, that of dividing the power on two units instead of, as then, on three. A few modifications have been introduced for certain constants used to agree with more recent observations.

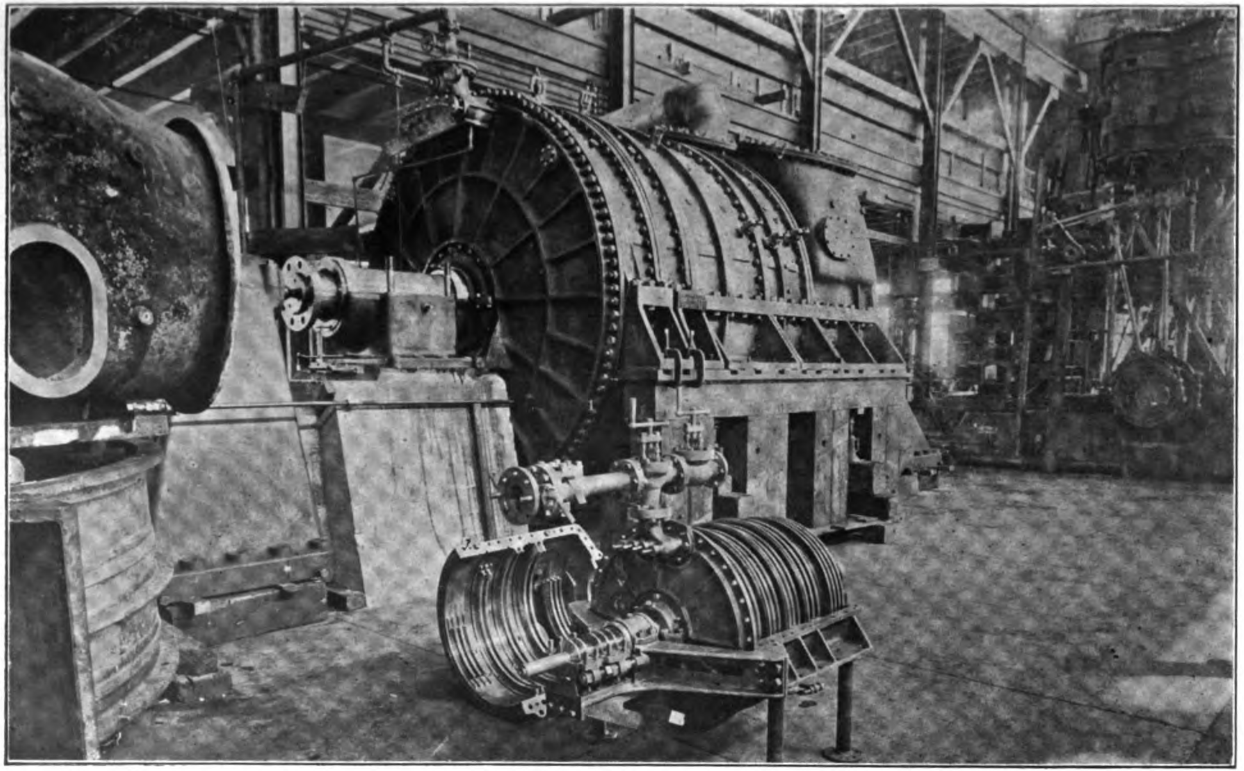
Preliminary design.—In making first approximations of revolutions of the turbine rotor we will base assumptions of vane speeds on knowledge gained in experiments carried out by those interested in the Curtis marine turbine. Suitable vane speeds vary between 175 and 190 feet per second with a turbine working within a pressure range of about 265 pounds absolute and with steam speeds from 2,000 feet per second down to 1,200 feet as obtained by expansion in various sets of nozzles arranged in a sufficient number of stages.

For reasons of more ready applicability it seems convenient to refer the problem to some specific case. Having this in view we will presume the following:

Type of ship, torpedo-boat destroyer.

Speed of ship, 33 knots per hour.

Effective horsepower required at this speed, 8,000, found by tank experiments on models.



Finished Marine Turbines. 8,000 and 250 H.P., Respectively.

Twin screws.

Curtis turbine, marine type.

Moreover, that the weight of engines for this class of machinery must not exceed 15 pounds per horsepower. Based on weights of turbines built, we find the mean vane diameter should be about 83 inches. Then, using an adopted mean vane speed of 183 feet per second, we get for revolutions of the rotor,

$$\text{R.P.M.} = \frac{60 \times 183}{\frac{83}{12} \times \pi} = 508,$$

which we will use to determine the dimensions of the propeller.

THE PROPELLER.

The great advantage of rotor speeds as high as condition will permit has been pointed out, and are a necessity both for efficiency in working and to minimize dimensions of the turbine. The propellers being driven by direct connection to the turbine rotor must revolve at the same high speed. The speed of revolutions and the diameters needed for certain powers bring the tip speed in many instances as high as 13,000 to 14,000 feet per minute, but although the efficiency of the propeller rapidly diminishes at such speeds, it has been found that the combined efficiency becomes greater and that the increased turbine efficiency more than counterbalances the decrease in propeller efficiency. A limit of 12,400 feet per minute for tip speed will be set for the example to follow.

Although many recent investigations and experiments have thrown light, as to design of propellers, on certain features as desirable and others as detrimental, the blade theory, propounded by Froude and worked out by Taylor for practical application (see Resistance of Ship and Screw Propellers, by D. W. Taylor), may be used with advantage in calculating the screw. This is especially true for a case as the one in hand dealing with conditions entirely different from those existent in engine work, and for which not a great deal of actual data are to be found.

Dimensions of propellers are influenced directly by—

1. Speed (K) of ship in knots per hour.
2. The effective horsepower ($E.H.P.$) needed for propulsion.
3. Real slip (S).
4. Diameter ratio $\left(\frac{D}{P}\right)$ which is the relation of diameter and pitch.
5. Number (N) of blades.
6. Pressure (Ps) per unit of projected blade surface.

Real slip of propeller $S = S_1\ (1-w) + w$,
in which

S_1 = apparent slip.
w = wake.

Pitch (P) and revolutions (R) of propeller must conform to assumed slip and speed, and the diameter ratio must be such as to permit the propeller to be run at the stipulated turbine revolutions.

The effective horsepower delivered by a propeller per second is obtained from

$$E.H.P. = \frac{1}{550 \times 60^3} P^3 \times R^3 \times D^2 \times N(I-S)(aSA-fB) \,. \quad (1)$$

The total horsepower absorbed by a propeller per second from

$$H.P. = \frac{1}{550 \times 60^3} P^3 \times R^3 \times D^2 \times N\,(aSA-fC), \quad . \quad (2)$$

Dividing (1) by (2) we get the propeller efficiency (e),

$$e = (1-S)\,\frac{aSA-fB}{aSA+fC}.$$

Applied to our example we get:

$E.H.P.$ = 8,000 to be equally divided among two shafts;

R = 508, determined by minimum speed of revolution per minute of turbine rotor with reference to weight and space, and assumed vane and steam speeds;

Speed of ship = 33 knots per hour;

Real slip = 8.5 per cent. from an assumed apparent slip = 4 per cent., and wake = 4.5 per cent;

Effective thrust, per screw,

$$T = \frac{E.H.P.}{K} \times 325.8 = \frac{\frac{1}{2} \times 8{,}000 \times 325.8}{33} = 40{,}000 \text{ pounds.}$$

Helicoidal area = (Disc — Hub) × disc area ratio.

The hub is approximated from shaft, easily obtained when power and revolutions are known.

Assume disc area ratio = .5 = M;

Tip speed = 12,400 feet per minute = T_s;

From drawing area of hub = 2.18 square feet = H.

Helicoidal area either by trial or some such simple formulae as this: $\left\{ \frac{I}{12.5}\left(\frac{T_s}{R}\right)^2 - H \right\} \times M = \left(\frac{12{,}400^2}{12.5 \times 508^2} - 2.18\right) \times .5$

$= (47.7 - 2.18) \times .5 = 22.8$ square feet.

The disc being 47.7 square feet gives a diameter of 7 feet 9½ inches. Having found the diameter corresponding to tip speed stipulated, and the area ratio to give certain allowable pressures, it remains to determine pitch for the assumed slip and to verify diameter for horsepower to be transmitted. We have $P \times R = \frac{101.33 \times 33}{1 - .04} = 3{,}483$, and, therefore, pitch (P) $= \frac{3{,}483}{508} = 6.85$ feet.

An approximate propeller is now laid down, Figure 1, and the blades sketched in, the area of which should be about 23 square feet. We will choose a three-bladed screw as most suitable.

The blade as drawn shows a maximum width ratio $\left(\frac{l}{D}\right)$ of about .35. From the approximately-found diameter and pitch we get

$$\text{Diameter ratio} = \frac{7.8}{6.85} = 1.14.$$

The blade characteristics A, B and C may now be found for this diameter ratio, and corresponding to the dimensions and contour actually existing in the blade drawn, by integrating functions involving diameter ratio, slip and width ratio.

(See Naval Architecture by Peabody, in which the procedure has been carried out for the "standard" blades. Any other blade must, of course, be treated separately.)

This being done, we get

$A = .878$, $B = 1.186$, and $C = 6.22$, which when reduced to actual width ratio (.35) gives

$A = .876 \times .35 = .3067$;

$B = 1.186 \times .35 = .415$;

$C = 6.22 \times .35 = 2.18$.

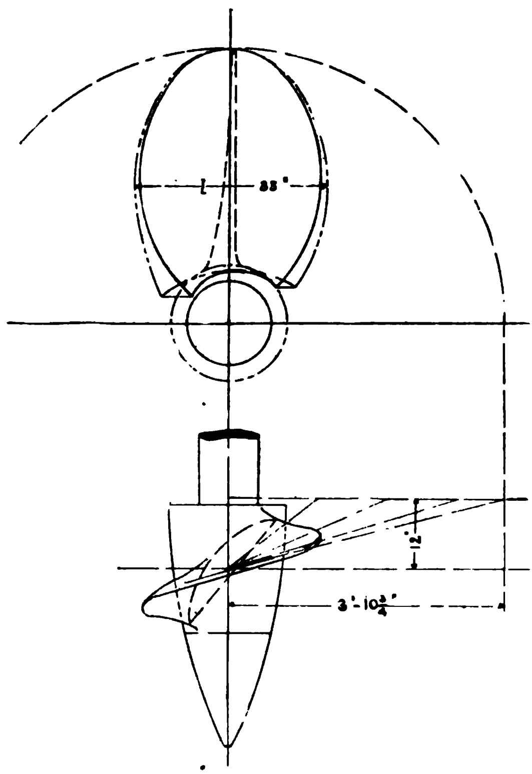


Figure 1.

With respect to other quantities we have for the thrust factor

$$a = 3.3 - .4\frac{D}{P} = 2.84, \text{ and}$$

f, co-efficient of friction, $= .016$.

$N = 3$ by choosing a three-bladed screw, and
$N\ (1-S)\ (aSA-fB) =$
$3\ (1-.085)\ (2.84 \times .085 \times .3067 - .016 \times .415) = .185.$

$$D = \sqrt{\frac{4{,}000 \times 550 \times 60^3}{3483^3 \times .185}} = \sqrt{60.7} = 7.8 \text{ feet.}$$

$$\text{Efficiency } e = (1 - .085)\frac{.0674}{.074 + .03488} = 57 \text{ per cent.}$$

The projected area from drawing = 19.7 square feet, and pressure per square inch of projected surface $P_s = \frac{40{,}000}{19.7 \times 144}$ = 14 pounds.

Total brake horsepower at turbine end of shafting = $\frac{1.04 \times 4{,}000}{.57 \times .98} = 7{,}460$, based on 57 per cent. propeller efficiency and .98 per cent. line-shaft efficiency, and 4 per cent. increase due to thrust deduction and wake.

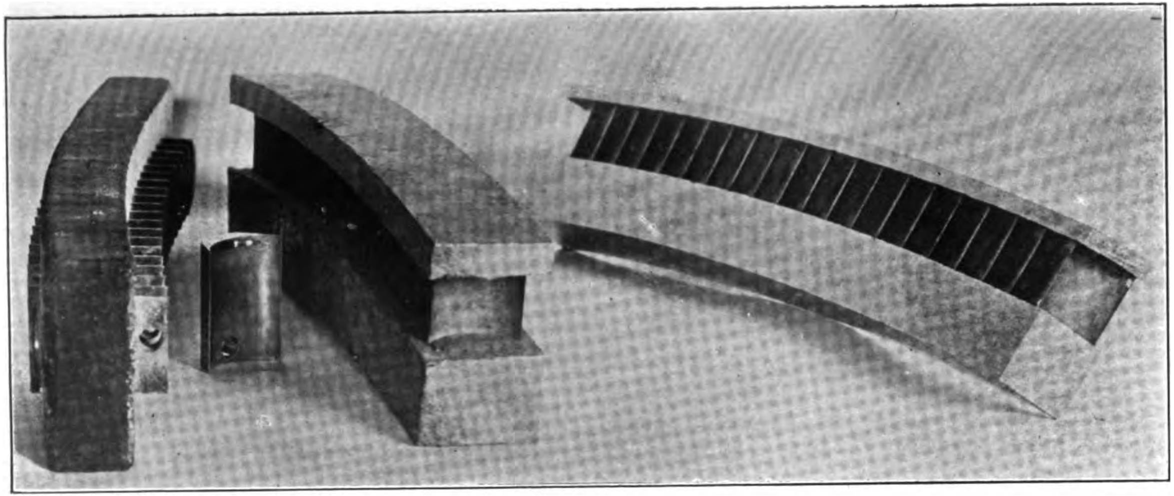
The indicated horsepower of a reciprocating engine to deliver foregoing effective horsepower would be $\frac{7{,}460}{.92} = 8{,}108$, based on 92 per cent. mechanical efficiency.

THE TURBINE.

Description.—The Curtis marine turbine was developed from the stationary type of turbine and contains all the essential features of that type. The first marine turbine built of Curtis type was for the yacht *Revolution*, of about 2,000 horsepower distributed on two shafts, each with one propeller. Each turbine had then only two ahead stages and one reversing stage arranged on the outer rim of the low-pressure rotor. The present marine turbine consists of seven ahead stages and two for backing, the rotors for each being all independently attached on the spindle or main shaft. The general construction as well as principal details may be understood by the description following and by reference to Figure 2, representing a marine turbine of about 3,500 B.H.P. running at 475 R.P.M.

The term "stage" implies the space within two diaphragms,

containing rotating and stationary vanes and a set of nozzles. The casing (A), made of cast iron, is built up of several sections of semi cylinders bolted together, the upper half being secured to the lower by longitudinal flanges. Dished heads (B), in one casting, are bolted to the ends of the cylinder and contain the shaft stuffing boxes (C). The packing in these stuffing boxes consists of double sectional rings of pure carbon, the space between being steam packed. The main shaft (D) is of wrought-steel and hollow, necessarily of a large diameter to secure rigidity. The rotors (E) are built up of cast hubs, forged rims and boiler-plate sides rivetted and screwed. The diaphragms (F) consist of cast or dished plates rivetted to steel rings, the inner of which has a composition boss within which the rotor hub revolves, the outer one a dovetailed extension fitting a groove in the casing, which holds each diaphragm in position. They are made in one piece, as there exists no need for removal; the upper casing, however, can be lifted up by disconnecting the flanges. The blading for each stage is made up of stationary vanes (G) and rotating vanes (H), the former in grooves of sectional attachments screwed to the inside of the casing, the latter in grooves of the rotor rim. The vanes are not, as in most other turbines, inserted and fastened independently or individually, but are made up in blade segments from 10 to 12 inches long, consisting of a foundation ring, the vanes and a shroud band all cast together. When the vanes are short, they may be milled out of the solid, but are usually built up as described, the vane itself being extruded from blooms of a special composition. The bodies (I), containing the nozzles, are secured to the casing at each stage, and consist of a casting with apertures separated by properly formed nickel-steel plates cast in the body. The steam inlet for the ahead stage is at (K) and for the backing stage at (L), the exhaust nozzle for both being at (M). Of the nozzles those in the first stage, both ahead and backing, are expanding, all the others are parallel. The rotor shaft rests in the bearings (N) and revolves in oil under pressure. Steam leakage outward in the stuffing boxes



BLADE SEGMENT.

A Blades in core ready for casting.
B Blade.
C Segment, rough casting.
D Segment, finished casting.

(or air leakage inwards) is brought to a minimum by connecting the annular spaces, between the carbon packing rings of the gland, to some intermediary stage where the pressure is only slightly above the atmosphere.

The steam velocities through the nozzles in the various stages are so arranged as to allot one-quarter of the total energy in the steam to the first stage and one-eighth to each of the other six. This is done to obviate an undesirably high shell pressure in the first stage, in which otherwise both the bursting and distorting stress would act detrimentally on the large shell area. Losses on account of the high frictional resistance, which occurs when a fast-rotating disc revolves in dense steam, are also, thereby, materially lessened. To illustrate the working conditions of a turbine of this kind the following data are given, which are supposed to answer to a turbine developing about 1,200 horsepower. The pressures put down in the third column, which are the stage pressures, convey clearly the situation. For any other turbine of different power, but working under similar conditions, the corresponding pressures will, of course, be nearly the same.

Table I.

Stage.	Absolute inlet pressure, pounds per square inch.	Absolute shell pressure, pounds per square inch.	Throat pressure in nozzle, pounds per square inch.	Square inch of throat required, one nozzle.	Absolute nozzle end pressure in lbs. per square inch.	Square inches of nozzle end required, one nozzle.
1	265.0	79.0	152.9	1.339	95.7	1.5048
2	79.0	41.7	45.58	4.24	45.58	4.24
3	41.7	21.2	24.0	7.8	24.0	7.8
4	21.2	10.4	12.23	14.84	12.23	14.84
5	10.4	4.9	6.0	29.25	6.0	29.25
6	4.9	2.2	2.83	60.66	2.83	60.66
7	2.2	1.0	1.27	135.0	1.27	135.0

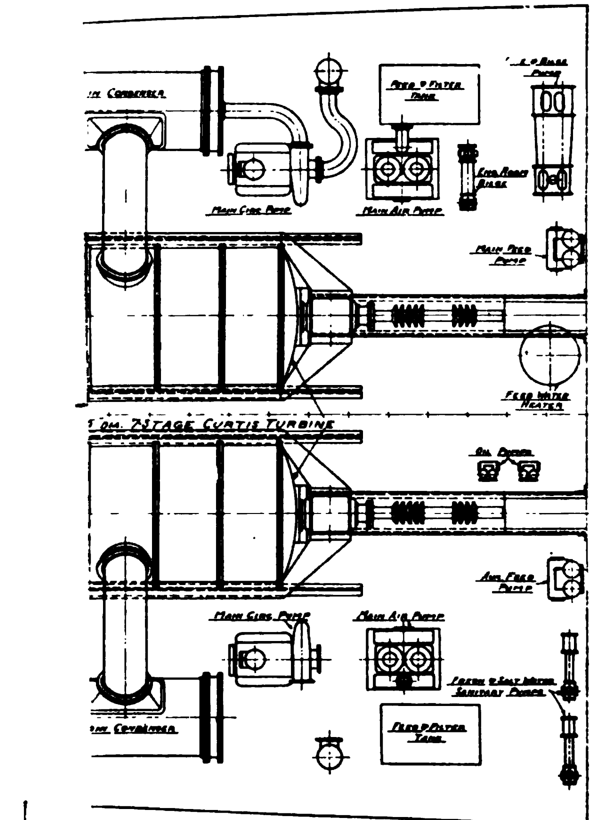
The first-stage nozzles have each a separately-operated disc valve for shutting off or opening to steam. All of the other stages are furnished with slide valves by means of which a certain number of their nozzles can be closed.

The steam flow through the turbine may accordingly be regulated to correspond with an amount of steam required for any desired horsepower, and the function of these valves thus bear the same relation to an economical variation of the power as do cruising turbines with Parsons' system. Regulating valves as described are necessary on turbines for naval ships cruising the greater part of their time at low power. For ordinary service the regulation may be accomplished by the throttle, and those valves are, therefore, limited on mercantile turbines.

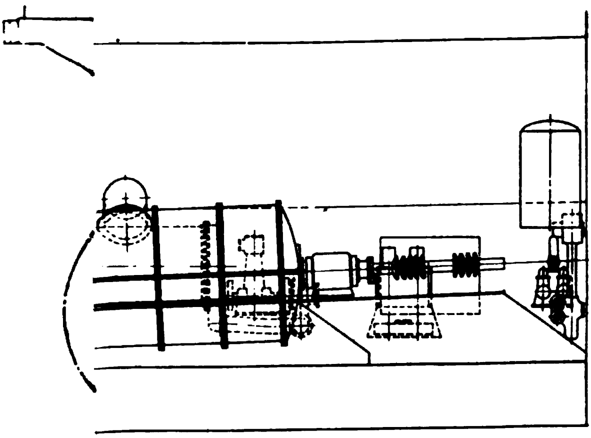
Besides the independently-operated stage valves pressure gauges are located on the shell to ascertain the pressure within each stage. A drain valve connects each stage in such a way as to lead the drain from a preceding stage direct into the next following and successively into the exhaust cavity, at which point a connection is made with the condenser.

Arrangement.—Machinery installations with Curtis turbine for propelling purposes of large power, as mentioned before, are arranged for two shafts in the ordinary way, see Figure 3. The disposition of the auxiliary machinery is, in general, also similar to that of arrangements with reciprocating engines. Two separate steam inlets are, however, required, one for ahead the other for backing, the pipes of which, with their throttle valves, connect to the nozzle bowls of the first stage. As there exists only slight end thrust on the rotor from uneven steam balance on the vanes within the cylinder, nearly all of the propeller thrust must be provided for in the usual manner in form of a thrust block, which is placed either forward or aft. The vane clearances being comparatively liberal all around, and there being an absence of dummy pistons with their close settings, renders micrometer adjustments in the thrust block entirely unnecessary. Gauges are, however, provided at the front and back of the turbine to enable the precise location of the rotor, with reference to the stationary parts, being determined.

Design.—The real difficulty when starting out in designing a steam turbine centers around the ability to determine the



PLAN



ELEVATION

efficiency, or, in other words, to closely approximate the pounds of steam needed per horsepower for given conditions of pressure at entrance and exhaust and defined stipulations as to the quality of the steam. To do this in any turbine, and especially in turbines of the impulse type, like Curtis or DeLaval, it is necessary to base one's assumptions on data secured in tests or by actual experiments. Such tests have been performed most carefully with the present seven-stage Curtis marine type. An average result of such tests justifies us to assume a steam consumption of 14 pounds per brake horsepower, using dry saturated steam at an initial pressure of 265 pounds absolute and a vacuum of 28 inches, the rotor vane speed at the same time to be approximately between 180 and 190 feet per second and the allotment of power as previously stated among the seven stages.

With the use of superheated steam, for which this type of turbine is particularly well adapted, the consumption will undoubtedly be considerably lowered, down to 13 pounds or better, which has been amply verified in various stationary turbine plants.

The total available heat per pound in steam expanding adiabatically between 265 pounds and 1 pound absolute determined by the formulae $H_1 - H_2 = Q_1 - Q_2 + H_v - T_2(E_{q1} + E_v - E_{q2})$ we find =

$380.2 - 70.0 + 825.6 - 563(.5728 + .952 - .1329) = 352.2$ B.T.U. (See J. A. S. N. E. of August, 1906.)

If all of the heat contained in the steam could be converted into work on the shaft the consumption of the ideal turbine would be $\dfrac{1{,}980{,}000}{352.2 \times 778} = 7.21$ pounds per horsepower-hour.

The actual turbine, on the basis of 14.0 pounds steam per horsepower-hour, will accordingly give an efficiency $= \dfrac{7.21}{14.0}$ $= 51.52$ per cent.

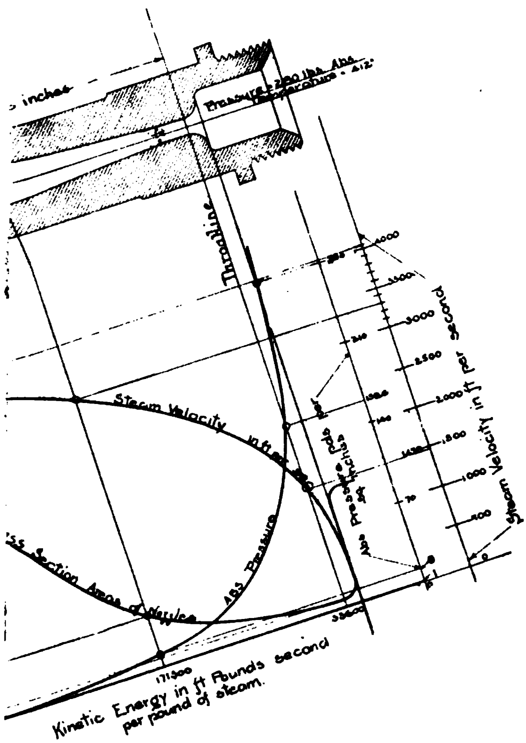
The losses in an impulse turbine may be attributed to the following causes:

1. Steam friction in nozzles.
2. Steam friction in vanes.
3. Steam shock on the vanes caused by deviation in the relative entrance velocities and by starting the quiescent steam between the vanes when passing nozzles.
4. Resistance to the revolving surfaces in steam of more or less density.
5. Radiation of heat from the cylinder, spreading and leakage of steam between vanes, nozzles and shaft bushings.
6. Loss of energy due to the exit velocity.
7. Mechanical losses, such as friction of journals and stuffing boxes.

The nozzle.—Weight, space and propeller speed renders it necessary, in marine turbine installations, to limit the rotor diameter as well as its speed of revolutions, and incidentally to use low steam speeds. These are obtained by arranging the turbine for a comparatively large number of "expansions" or stages and by using appropriate types of nozzles.

The general principles of steam expansion and the velocity attained by the jet in different nozzles has been written up extensively in various treatises and does not come within the scope of this paper. Suffice it to say, therefore, that the jet velocity depends essentially upon the "ratio of expansion," or on the proportions observed in the areas of throat and outlet end of the nozzle and upon the initial and terminal pressure at the orifice and outlet. Moreover, that a definite relation always exists between the initial pressure and that of the throat, when the orifice is well rounded. Thus if P signifies the initial pressure, the throat pressure will be .577 P if the outlet pressure is equal to this or any other pressure below .577 P down to a perfect vacuum. This occurs in either a parallel or an expanding nozzle.

The diagram appended, Figure 4, of a DeLaval conical divergent or expanding nozzle, calculated to expand steam from 280 pounds absolute to 1 pound absolute, illustrates the relation of pressures, cross-sectional areas and velocities at different parts of the nozzle. The energy resulting from



inches
Throat line
Steam Velocity in ft per second
Abs Pressure lbs per sq inches
ABS Pressure
Section Area of Nozzle
171500
35600
Kinetic Energy in ft Pounds second per pound of steam.
4000
3500
3000
2500
2000
1500
1000
500
0

the ideal velocities, the assumed frictional losses and consequent losses in velocity both in nozzle and buckets of a single-impulse wheel, and the resultant efficiencies, are also diagrammatically shown. Judging from the extremely high velocity attained, it may be readily inferred that even with a small ratio of expansion, or a very slight increase of outlet area over throat area, a considerable velocity results. For pressures below 70 pounds per square inch the parallel nozzle has been found more efficient than the expanding, and is used in all stages, except the first, of a Curtis marine turbine. The steam weight discharged is directly due to the throat velocity and the area at the throat as well as the specific volume, therefore the higher the initial steam pressure the greater the weight per unit of time and area. The energy of the jet, however, corresponds to the end velocity as well as the weight. Throat and end velocity is the same in parallel nozzles, but quite different in the expanding nozzle.

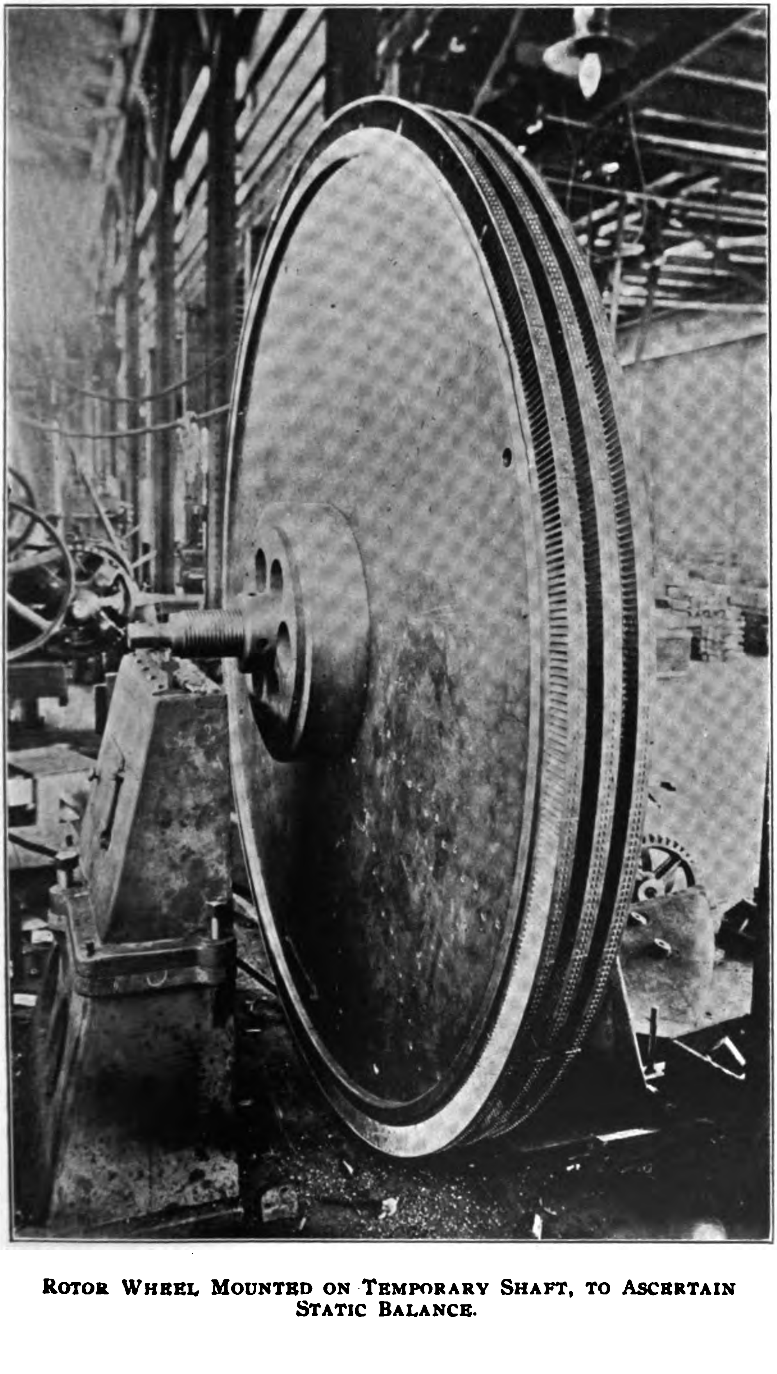
Action.—The Curtis marine turbine is of the compound-impulse type. The steam expands in seven sets of nozzles or pressure stages successively from the initial pressure to that of the exhaust. The first pressure stage, as has been mentioned, develops one-fourth the total energy, the other six each one-eighth. The jet energy is transformed into work by impulse on the moving discs, which, according to the jet velocity used, are arranged in from three to four velocity stages. Good efficiency demands that certain proportions of steam and bucket velocity be observed throughout the turbine.

The steam action is essentially as follows: After expansion in the nozzles of the first stage the steam issues in solid jets against the first row of moving buckets, which absorb a part of the jet energy, and, after passing through said buckets, meets the first row of stationary vanes. The purpose of these vanes is to guide the steam into the second row of moving buckets, which, in their turn, take up another portion of the kinetic energy still possessed by the fast flowing steam, at the same time diverting it into the second row of stationary buckets, which deflect the steam on the third row of moving buckets,

where again more of the energy is taken up. In the first stage, where the initial nozzle velocity is considerably higher than in any subsequent stage, this operation is again repeated in a third stationary row and a fourth moving row, the remaining energy after that being too small to warrant additional buckets. The pressure at the outlet end of the nozzles is brought down to the pressure within each stage by slight expansion in the various rows of buckets, the volume of the steam corresponding to successive pressures. However, due to the fact that the velocity of the steam is gradually diminished by the continuous absorption of energy, the passages traversed by the steam must be enlarged. This is provided for by lengthening the buckets as well as by increasing the vane angles in each succeeding row.

After leaving the last row of moving buckets in each stage the steam attains partial rest before it enters the nozzles of the next stage, in a manner similar to that which occurs between each "expansion" of a Parsons turbine. On entering the nozzles of the second stage the steam again expands, whereby new velocity is given, and now acts in the various rows of buckets of that stage exactly as it did in the first stage, and so on right through all of the seven stages of the turbine. There is this difference, however, that, owing to the nozzle velocity being very much less in the stages succeeding the first, three moving and two stationary rows will suffice there instead of respectively four and three of the first stage. The number and size of the nozzles in the different stages must obviously conform to the velocity and the volume of the steam as a result of expansion through the turbine. Due to this fact we find the nozzles circumscribing only a small arc in the first stage, gradually increasing in the following, until, in the last stage, the entire circle is completely filled with nozzles. This latter condition, however, is governed wholly by the power in comparison with the rotor diameter.

Steam velocities through the buckets of the first stage relatively to a fixed vane speed are shown diagrammatically in Figure 5. No account has been taken of velocity increase



Rotor Wheel Mounted on Temporary Shaft, to Ascertain Static Balance.

as a result of expansion in the buckets in this diagram, which must be done in figuring vane dimensions. In the diagram shown in Figure 7, laid out for the seventh stage, this change is observed.

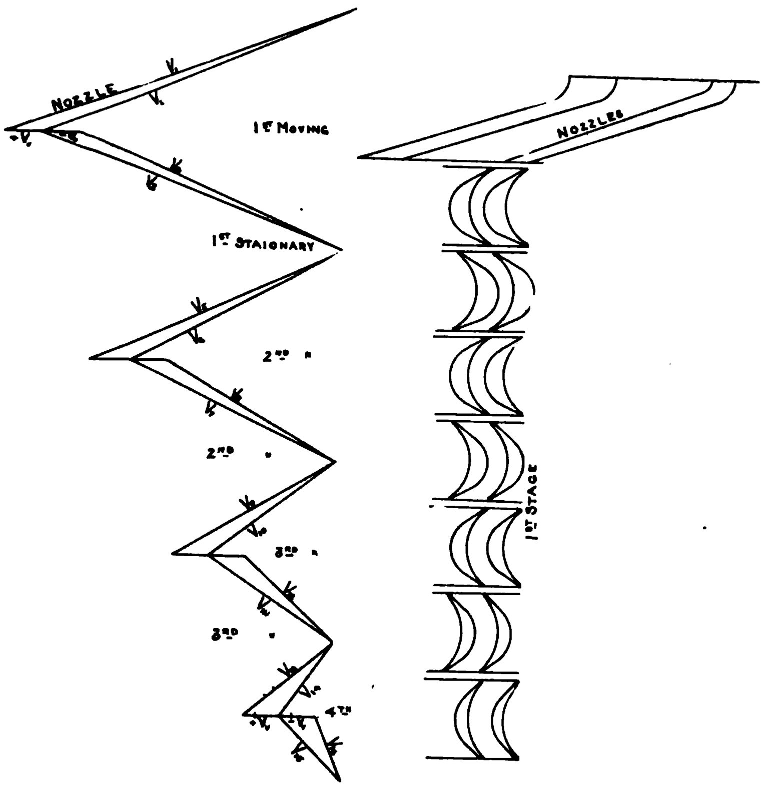


Figure 5.

VELOCITY DIAGRAM, 1ST STAGE. EXPANSION WITHIN BUCKETS NOT FIGURED.

Calculation.—In a steam turbine, the potential energy of the admitted steam is converted into kinetic energy of the steam jet and the kinetic energy of the steam jet into impulse force on the revolving wheel. This double transformation can take place either simultaneously in the same wheel (reaction turbine), or successively in stationary nozzles and revolving wheels (action or impulse turbine).

In the reaction turbine, steam enters the revolving wheel at admission pressure at moderate velocity, expands in the wheel and leaves the wheel at the velocity of expansion relatively to the wheel, that is, in a vane completely reversing the jet, at an absolute velocity equal to expansion velocity minus wheel velocity. To take the full energy out of the steam, it must leave the wheel at zero absolute velocity; that is, the wheel velocity must equal the velocity of the steam jet.

In the impulse turbine, the steam expands in a separate nozzle, strikes the rotating wheel at expansion pressure and expansion velocity, and rebounds therefrom, as an elastic body, with the same velocity, minus the friction loss, as that with which it strikes, hence losing velocity by an amount equal to twice the wheel velocity. To take the full energy out of the steam, the wheel velocity must therefore equal one-half the velocity of the steam jet.

In an impulse turbine, therefore, such as the Curtis turbine, with a given pressure range per wheel, the wheel velocity at best efficiency is only about 50 per cent. of what it is in a reaction turbine, and $\frac{1}{\sqrt{2}} = 70$ per cent. of what it is in a combination turbine alternating between impulse and reaction effect, such as the Parsons turbine.

Since the most difficult condition of turbine design is the high rotor velocity required to utilize the high steam velocity, the impulse turbine is theoretically superior. It is not possible to get efficiency with a single turbine wheel of the reaction type, while with a single wheel of the impulse type good efficiencies have been reached by using extremely high peripheral speeds (DeLaval turbine). In general, however, while hydraulic turbines are always designed with a single wheel, in the steam turbine it is not possible to make effective use of the total energy of the steam by a single wheel, but a number of wheels in series have to be used. That is, the rotor velocity is reduced by subdividing the total pressure range into a number of successive stages. In the combination impulse and reaction turbine of Parsons, each expansion or

pressure stage requires a revolving wheel and a stationary guide wheel, the steam acting by impulse at entrance and by reaction at the leaving edge.

In the impulse turbine, in which the expansion of the steam is carried out in nozzles separately from the vane system, a further effective means of speed reduction is presented by the use of several velocity steps in each stage; that is, by imparting the velocity of the current of steam to a number of successive wheels composing a single revolving disk, with stationary intermediate guide wheels.

Since in a revolving wheel of an impulse turbine the steam velocity is reduced by an amount equal to twice the wheel velocity for maximum efficieney, a single wheel per stage must revolve at one-half the steam velocity, two wheels per stage at one-quarter steam velocity, etc. The use of several wheels per stage, therefore, is a more effective means of reducing the rotor speed—or inversely, at given rotor speed, reducing the total number of revolving wheels—than the use of several expansion steps. A two-wheel stage can take care of twice the steam velocity (that is, four times the steam energy) of a single-wheel stage, and therefore replaces four single-wheel stages; or, in other words, the speed reduction of the rotor is proportional to the number of wheels per stage, that is, the number of velocity steps; on the other hand, it is proportional to the square root of the number of stages, that is, number of pressure steps.

The simultaneous use of pressure steps, or expansion stages, and velocity steps, or number of wheels per stage, therefore leads to a construction requiring a comparatively small total number of wheels, as carried out in the Curtis type of turbine.

The steam weight (W) passing (rate of flow) through a turbine nozzle depends principally upon:

1. Throat area $= A$ (in square inches),

2. Specific volume of the steam after expansion $= v$ (in cubic feet per pound),

3. Velocity of flow at the throat $= V$ (in feet per second), and is expressed by

$$W = \frac{A \times V}{v \times 144}, \quad \ldots\ldots\ldots \quad (1),$$

in which

$W =$ weight in pounds discharged per second.

The velocity at the throat corresponds to the thermo-dynamic head made up of the difference between the total heat of the entering steam and the total heat corresponding to the throat pressure, when the steam expands adiabatically. The throat pressure, for maximum flow, must be equal to, or less than $.577\ P_1$, when P_1 signifies the initial pressure. Owing to steam friction in the nozzle the velocity becomes somewhat less than if no diminution of heat conversion occurred, and must be carefully taken note of. We will call the percentage loss due to friction y, and if the equivalent thermo-dynamic head is called H, we get $(1-y)\ H$ representing the head creating kinetic energy.

H is most readily obtained by aid of the ordinary entropy-temperature diagram, shown simply in Figure 6.

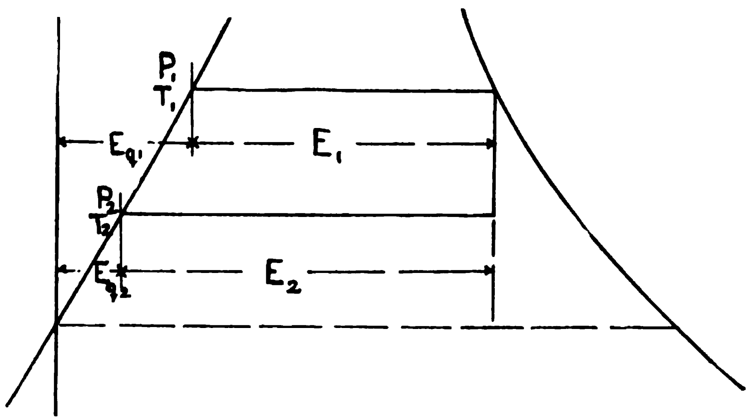


Figure 6.

We make

$P_1 =$ initial pressure in pounds per square inch absolute;

$T_1 =$ absolute temperature in degrees Fahrenheit corresponding to P_1;

$E_1 =$ entropy corresponding to P_1 or T_1;

$P_2 =$ throat pressure in pounds absolute $= .577\ P_1$;

T_2 = absolute temperature corresponding to P_2;

E_2 = entropy corresponding to P_2 or T_2.

The quantities for entropy are obtained direct from the steam tables, thus:

$E_1 = \frac{H_v}{T_1}$ when H_v = heat of vaporization at T_1.

$E_2 = E_{q1} + E_1 - E_{q2}$, when E_{q1} and E_{q2} = entropy of the liquid respectively at T_1 and T_2.

We have the general formulae for the relation between kinetic energy and heat expressed by

$\frac{V^2}{2g} = 778\,(H_1 - H_2)$, in which

V = velocity in feet per second,

$g = 32.16$.

H_1 = total heat of steam at P_1.

H_2 = total heat of steam at P_2.

The entropy curve being nearly a straight line, sufficiently close results will be obtained, for finding the available heat producing flow, by taking "mean entropy." We then get, observing friction,

$$V = 158\sqrt{(E_1+E_2)\,(T_1 - T_2)\,(1 - y)}, \quad . \quad . \quad (2)$$

which may be used for finding the velocity in any part of the nozzle.

Example:

Initial pressure = 265 pounds abs. = P_1; T_1 = 867° F.,

Throat pressure = .577 P_1 = 152.8 pounds abs. = P_2; T_2 = 820° F.,

$E_1 = .9524$,

$E_2 = 1.01$,

y, disregarded,

$V = 158\sqrt{1.9624 \times 47} = 1{,}517$ feet per second.

The steam weight passed, when

$A = .6156$ square inches,

$v = 2.96 \times \frac{1.01}{1.05} = 2.8416$ cubic feet, is

$W = \frac{.6156 \times 1517}{2.8416 \times 144} = 2.283$ pounds per second.

The steam velocity found above refers strictly to conditions of expansion set forth, and is attained at the throat of whatever kind of nozzle is used, divergent of parallel. For the parallel nozzle, in which the throat and outlet area are identical, the throat velocity and outlet velocity will be the same, and this velocity can, therefore, be used in finding the energy of the jet. In the divergent nozzle a farther expansion occurs within the nozzle between the throat and the outlet, and the velocity at the outlet, which determines the energy, corresponds to the total heat drop taking place between the orifice and the outlet. Thus, substituting for the throat pressure the pressure at the outlet in the first stage nozzles given in table 1, we get for V, disregarding friction,

$V = 158 \sqrt{(.9524 + 1.056)\ (867 - 784)} = 2{,}040$ feet per second, and when taking account of friction, assuming $y =$ 15 per cent,

$V_1 = 158 \sqrt{167\ (1 - .15)} = 1{,}881$. (See Figure 6.)

Tabulating the different quantities for the nozzles given in Table I, and figuring first, corresponding to H, disregarding friction, and second corresponding to H^1 assuming a certain value for y, we get

TABLE II.

E_1+E_2	T_1-T_2	H	V	H^1	V_1	Y	No. of stage.
2.0086	83	167.0	2,040	142.0	1,881	.15	1
2.3382	36	84.16	1,454	72.4	1,350	.14	2
2.579	32	82.6	1,435	71.06	1,332	.14	3
2.794	29	81.0	1,422	69.7	1,320	.14	4
3.020	24	72.48	1,345	62.5	1,250	.14	5
3.304	22	72.68	1,345	62.5	1,250	.14	6
3.498	20	70.0	1,324	60.2	1,230	.14	7
1	2	3	4	5	6	7	

The cross-section of the nozzles used in Curtis turbines is rectangular, the expansion being obtained by the divergence of the walls of the nozzle. The energy developed per pound of steam per second in each, and corresponding to foregoing velocities, is:

1st stage $\frac{1881^2}{64.32} = 55{,}000$ foot pounds,

2d stage $\frac{1350^2}{64.32} = 38{,}320$ foot pounds,

3d stage $\frac{1332^2}{64.32} = 27{,}600$ foot pounds,

4th stage $\frac{1320^2}{64.32} = 27{,}080$ foot pounds,

5th stage $\frac{1250^2}{64.32} = 24{,}280$ foot pounds,

6th stage $\frac{1250^2}{64.32} = 24{,}280$ foot pounds,

7th stage $\frac{1230^2}{64.32} = 23{,}520$ foot pounds.

Total, 220,000 foot pounds.

Assuming a uniform friction loss of eight per cent. in the vanes, then $y = .08$ and $V_x = V_e \sqrt{1-y}$, the following velocities in each vane row are obtained, reference being had to the velocity diagram shown in Figure 5, V_x and V_e in the formulae signifying respectively exit and entrance velocity from and to the vanes. No account of the increase in velocity due to expansion within the vanes being observed.

$V_1 = 1881$, nozzle velocity, absolute entrance,
$V_2 = 1708$, nozzle (1_m) velocity, relative entrance,
$V_3 = 1637$, vane (1_m) velocity, relative exit,
$V_4 = 1464$, vane (1_s) velocity, relative entrance,
$V_5 = 1403$, vane (1_s) velocity, relative exit,
$V_6 = 1235$, vane (2_m) velocity, relative entrance,
$V_7 = 1184$, vane (2_m) velocity, relative exit,
$V_8 = 1017$, vane (2_s) velocity, relative entrance,
$V_9 = 975$, vane (2_s) velocity, relative exit,
$V_{10} = 815$, vane (3_m) velocity, relative entrance,
$V_{11} = 781$, vane (3_m) velocity, relative exit,
$V_{12} = 637$, vane (3_s) velocity, relative entrance,

$V_{13} =$ 611, vane (3_s) velocity, relative exit,
$V_{14} =$ 480, vane (4_m) velocity, relative entrance,
$V_{15} =$ 460, vane (4_m) velocity, relative exit,
$V_{16} =$ 350, vane (4_m) velocity, absolute exit,
$V_v =$ 183, angular vane speed.
All of foregoing velocities are in feet per second.

The kinetic energy loss with y per cent. friction loss will be $\frac{yV_r^2}{2g}$ when V_r is the relative entrance velocity in a moving row of vanes. Moreover, if the steam leaves the same row with an absolute velocity V_a its energy is $\frac{V_a^2}{2g}$. Subtracting both losses from the energy caused by the initial jet velocity V, we have, for the work done upon any one of the moving buckets

$$\frac{V^2}{2g} - \frac{yV_r^2}{2g} - \frac{V_a^2}{2g}.$$

Tabulated for foregoing velocities, the work done per pound of steam in the first stage will be in foot-pounds second:

First moving buckets,

$$\frac{1}{2g}\left\{ V_1^2 - .08V_2^2 - V_4^2 \right\} = \frac{1{,}881^2 - .08 \times 1{,}708^2 - 1{,}464^2}{64.32} =$$

17,930 foot pounds.

Second moving buckets,

$$\frac{1}{2g}\left\{ V_5^2 - .08V_6^2 - V_8^2 \right\} = \frac{1{,}403^2 - .08 \times 1{,}235^2 - 1{,}017^2}{64.32} =$$

12,600 foot pounds.

Third moving buckets,

$$\frac{1}{2g}\left\{ V_9^2 - .08V_{10}^2 - V_{12}^2 \right\} = \frac{975^2 - .08 \times 815^2 - 637^2}{64.32} =$$

7,644 foot pounds.

Fourth moving buckets,

$$\frac{1}{2g}\left\{ V_{13}^2 - .08V_{14}^2 - V_{16}^2 \right\} = \frac{611^2 - .08 \times 480^2 - 350^2}{64.32} =$$

3,570 foot pounds.

Total = 41,744 foot pounds.

Assuming (without detail figuring) the original proposition is realized, namely, that each subsequent stage following the first develops one-half the power of the first, we have for the work in the whole turbine converted into heat units:

1st stage =	41,744 foot pounds =	53.6 B.T.U.
2d stage =	20,872 foot pounds =	26.8 B.T.U.
3d stage =	20,872 foot pounds =	26.8 B.T.U.
4th stage =	20,872 foot pounds =	26.8 B.T.U.
5th stage =	20,872 foot pounds =	26.8 B.T.U.
6th stage =	20,872 foot pounds =	26.8 B.T.U.
7th stage =	20,872 foot pounds =	26.8 B.T.U.
Total all stages =	166,976 foot pounds =	214.5 B.T.U.

The losses throughout the turbine will then be about as follows:

From the efficiency given, or 51.52 per cent., the total losses = 48.48 per cent.

Friction loss in nozzles =	14.14 per cent.
buckets =	8.00 per cent.
Loss due to exit velocity =	9.53 per cent.
	31.67 per cent.

Deducting this from total losses, loss from all other sources will be 48.48—31.67 = 16.81 per cent.

Horsepower developed per pound of steam $\frac{167{,}000}{550} = 304.$

Steam quantity required for 7,460 horsepower $\frac{7{,}460}{304} =$ 24.54 pounds per second.

Steam per horsepower-hour $\frac{1{,}980{,}000}{167{,}000} = 11.84$ pounds.

Actual experiments with turbines of this type, (with, however, only three revolving rings of buckets in the first stage, and developing only about 1,200 horsepower) have demon-

strated that 14 pounds are needed, the difference, or 2.16 pounds, is therefore, the amount required to overcome other losses previously enumerated.

Energy by reheating of the steam on account of friction and the impact by the exit velocity is given back to a subsequent stage in form of heat by raising the quality of the steam. This amount may be accurately figured and the results obtained used in determining the blade dimensions, as well as the heat to be expended in the various stages.

Referring to Table I it will be noticed that the nozzle end pressure is higher than the shell pressure. In traversing the vane path a successive drop of pressure must, therefore, occur. The expansion taking place between the rows will in all likelihood somewhat mitigate the effect of friction and slightly increase the velocities. A difference of pressure on each side of the vanes over an arc covered by the stationary buckets does set in at the same time, whereby a slight end pressure on the rotor, opposite in direction to the propeller thrust, is exerted.

Determination of blade and nozzle dimensions.—For the example chosen the mean vane diameter was taken to be 83 inches, the horsepower of each turbine equivalent to 7,460 brake and the initial steam pressure about 265 pounds absolute, with 28 inches vacuum.

Since the horsepower primarily depends upon the steam quantity and the energy developed by its velocity, a sufficient number of nozzles, of dimensions such as to give the proper pressure range for each stage, must first be provided. For 7,460 horsepower at 14 pounds per hour, steam per second required equals 29 pounds.

Using velocities previously given for the first-stage nozzles and making $y = .15$, the area at the throat must be, formula (1),

$$A = \frac{29 \times 2.8416 \times 144}{1{,}517 \sqrt{1 - .15}} = 8.21 \text{ square inches,}$$

making each nozzle = .6156 square inches.

$$\text{Number of nozzles} = \frac{8.21}{.6156} = 13.3.$$

Turbines are generally made for considerable overload, even up to 50 per cent., which, if followed out here, would give about 20 nozzles.

The area of the nozzle end is figured for the same steam weight, but the velocity being considerably greater, or 1,881 feet against 1,491 feet, and the specific volume corresponding to the lower pressure and quality due to expansion being 4.294 against 2.841, we get for the nozzle end,

$$A_1 = \frac{29 \times 4.294 \times 144}{1,881} = 10 \text{ square inches, or for each}$$

$$= \frac{10}{13.3} = .76 \text{ square inches.}$$

The vane angles should be those given by the velocity diagram, one of which must be laid down separately for each stage. The passage area between the vanes must correspond with the velocity at each point as well as the actual volume. This must be figured for both quality incident to frictional disturbance and adiabatic expansion. A small amount is added to the length of the vane at each step over what is necessary for actual area in order to catch the steam separated from the jet in spreading and leakage.

The passage area between the vanes may be determined in the following manner. Owing to the fact that the volume is greatest after expansion in the last stage and the velocity is smallest at the last row, the buckets at this point become longest and, therefore, in a measure settle the outside diameter of the turbine. The dimensions of these buckets should be figured first on that account.

The velocity diagram, Figure 7, is laid down with the actual exit velocity from the nozzle as jet velocity. Referring to Table I, the pressure at the nozzle end is 1.27 pounds absolute and in the shell about 1 pound absolute, therefore a drop of .27 pound between the inlet of the first buckets and the outlet of the last. This pressure drop for adiabatic expansion is equivalent to 11 B.T.U., which we may assume is equally absorbed among the five rows, thus in each 2.2 B.T.U.

The relative entrance velocity to the first moving buckets

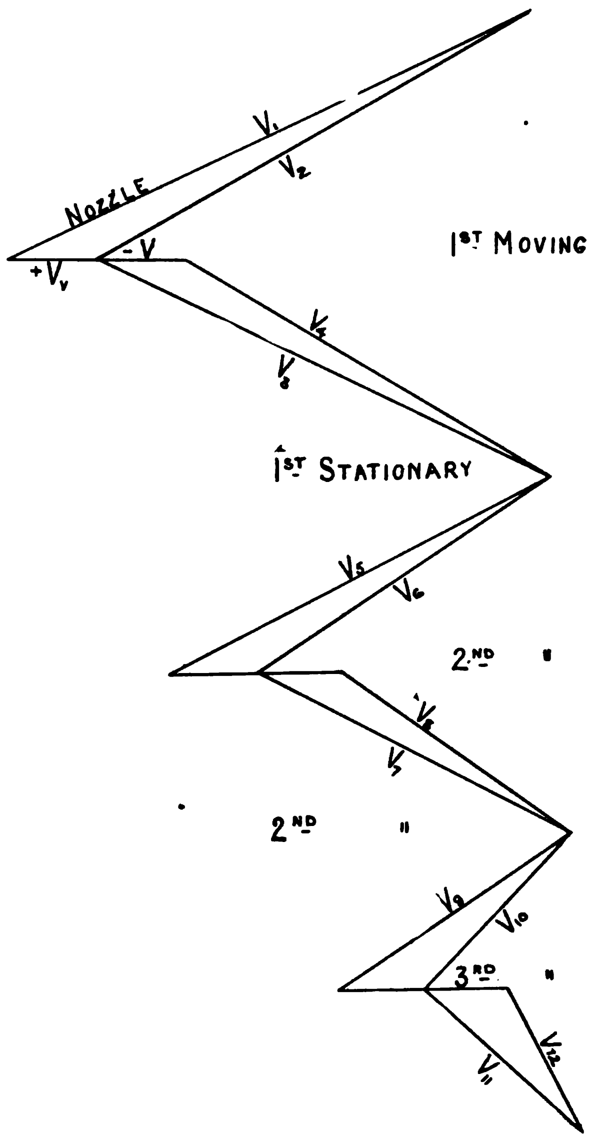


Figure 7.

VELOCITY DIAGRAM, 7TH STAGE, EXPANSION WITHIN BUCKETS OBSERVED.

is 1,065 feet from the diagram. Assuming a friction loss of 9 per cent. both in the energy of the steam passing the vanes and in the heat given up by the steam in transit, we get

$$\frac{V_3^2}{2g} = .91 \times 778 \times 2.2 \pm \frac{V_2^2}{2g} \times .91,$$

$V_3 = \sqrt{64.32 \times .91 \times 778 \times 2.2 + 1{,}065^2 \times .91} = 1{,}065$ feet per second.

In arranging for a slight steam expansion within the buckets, provided for by an excess of nozzle end pressure over shell pressure, an increased velocity is attained, rendering possible a reduction in their length with the same clear space between them as would be required otherwise with the longer bucket.

Vane speed being given and exit angle determined upon, V_4 is drawn in. The diagram is figured similarly right through for the remaining rows, whereby is found V_{11}, the relative exit velocity from the last row to be 460 feet. The pressure within the row at the point of smallest section equals about 1.016 pounds absolute.

Corresponding volume = 316 cubic feet.

Corresponding quality = .79.

Corresponding clear space between two buckets = .5365 inches.

Number of spaces $= \frac{261}{.66} = 400$; 261 = mean circumference, and .66 = space occupied by a bucket and an opening in inches.

Total steam volume $= .79 \times 316 \times 2g = 7{,}240$ cubic feet.

Area of steam flow $= \frac{7{,}240}{460} = 15.7$ square feet.

Length of blade $= \frac{15.7 \times 144}{400 \times .5365} = 10.14$ inches.

All blade or bucket passages may be determined in the same way, observing that the height of the first row must be made to conform with the nozzles.

Application.—Up to date no actual trials have been performed with the multiple-stage marine turbine of the modern Curtis type. A number of such turbines, however, are now

under construction, both for mercantile and naval purposes, most of which are now in a state of completion, such that early trials are anticipated. Among those alluded to, the following are of particular interest:

U. S. scout cruiser *Salem* of 16,000 horsepower, to make 24 knots with the propellers revolving at 350 turns per minute.

The Southern Pacific mail steamer *Creole* of 17 knots and 8,000 horsepower and a propeller speed of 235 revolutions.

Two sets of two units, each of about 25,000 horsepower combined, for the Japanese Navy.

A naval launch of about 250 horsepower.

All of the foregoing are being built by the Fore River Shipbuilding Company of Quincy, Mass.

The Hamburg-American liner *Kaiser*, equipped with Curtis turbines of a type adopted by the Allgemeine Electricitäts Gesellschaft of Berlin, has been in commission for some time, and is giving, it is believed, general satisfaction. Her tur-

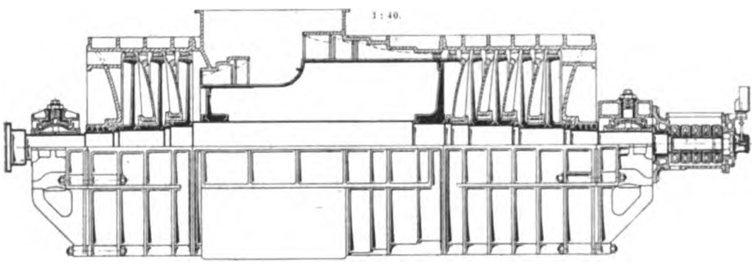


Figure 8.

bines consist, in the high-pressure zone, of five pressure stages, each with about three moving rows of buckets, while in the low-pressure zone there are some twenty pressure stages, each with a velocity stage or moving row. The reversing end of the turbine, figured for 50 per cent. of the full power, contains two pressure stages, each of which has three rows of moving buckets. Figure 8 shows a turbine in which is incorporated features of both Curtis and Parsons, while the *Kaiser's* turbines seem to possess features like the Zoelly.

The following items are especially interesting with respect

to the impulse type of turbine in review with the compound reaction type:

1. For a given bucket speed and pressure range the number of vane rows is considerably less in the former than in the latter.

2. The steam expansion being executed in nozzles renders possible the use of high initial pressures, as well as superheat, without affecting detrimentally the turbine cylinder from excessive stress, or the vane system from distortion incidental to superheat.

3. Aside from throttle-valve regulation the steam supply may be varied, to suit any degree of power to be developed by the turbine, by shut-off valves on the nozzles, eliminating thereby the necessity for separate cruising turbines as now usually arranged for in war ships.

4. Large clearances are used in Curtis marine turbine at both side and end of blades, thus minimizing the danger of fouling of buckets from displacement of shaft or vibratory influences.

5. Balance or dummy pistons, to equalize end pressure, do not exist in the impulse marine turbine, and a source of loss as well as the necessity for delicate adjustment is eliminated.

6. For the same total power propeller diameters are larger in twin-screw ships fitted with Curtis turbines than in vessels of triple or quadruple screws with Parsons. The capacity for maneuvering is increased by using larger propellers and by making available all of the blade area, against the two-thirds to one-half provided for in turbine ships of latter screw arrangement. Backing turbines are, however, fitted on each shaft in quadruple screw arrangements of naval ships.

It is needless to say that the trials of the Curtis marine turbine will be watched with keen interest, and, if the performance may be prejudged, success is predicted. Economy, natural advantage in application, saving of weight and space, should render this type of turbine especially serviceable.

SUGGESTIONS FOR THE CARE AND OPERATION OF NAVAL MACHINERY IN THE ENGINEER DEPARTMENT, U. S. NAVY.

By H. C. Dinger, Lieutenant, U. S. N., Member.

PART IV.

PIPING.

The various piping systems are among the chief items for care and repair on board ship. The following general information with regard to the latest practice of the Bureau of Steam Engineering is given.

PIPE FLANGES AND FITTINGS.

"The dimensions of all flanges will be in accordance with the Bureau of Steam Engineering table of flanges.

"There will be no screw joints in steel steam piping. The joints will be made by flanges of stamped or forged mild steel of the same quality as specified for steel steam pipes. The pipe will be either welded to the flange or else rolled into the flanges and beaded over to fit a recess flush with the face of the flange, in accordance with the practice of the Bureau of Steam Engineering. All flanges will be faced and grooved.

"Copper pipes will have flanges or approved composition couplings brazed on, and the end of the pipe will be beaded into a recess in the face of the flange.

"All copper pipes below the floor plates will have composition flanges.

"All joints between flanges in steam and exhaust pipes will be made with corrugated-copper gaskets.

"All joints between flanges in water pipes will be in accordance with Navy Department Standard No. 86, 'Method of

making joints and of obtaining water-tightness through bulkheads, decks, etc.'

"All flanges on the main drain connections to the main circulating pumps, and Construction and Repair drainage manifolds for connection with steam machinery, will be located to the satisfaction of the Bureau of Steam Engineering and the Bureau of Construction and Repair.

"Flanges for wrought-iron or steel pipes, for salt water, to be cast or wrought steel, galvanized.

"Fittings for steel steam pipes will be of cast steel, Class B, or composition.

"Fittings for copper pipes will be of composition unless otherwise directed.

"Fittings for wrought-iron or steel pipes, for salt water, to be of cast-iron or steel, galvanized.

"Pipes will be so led and flanges so placed that they may be readily taken down for renewal or repairs, and so that joints will be kept readily accessible.

"All external fittings on boilers will be composition unless otherwise directed, and will be flanged and through-bolted, or attached in other approved manner.

"All internal pipes will not touch the boiler plates anywhere, except where they connect with their external fittings. The internal feed and blow pipes will be expanded in the holes in boiler shells to fit the nipples on their valves, or will be secured in other approved manner, and will be supported where necessary and as directed.

"All internal pipes and fittings of brass, copper or composition will be tinned.

"All copper pipes intended to convey salt water will be protected on the inside by coats of asphaltum or other approved waterproof varnish, applied as hot as possible and in successive coats.

"All copper pipes in bilges will be well painted, and will not rest in contact with any of the iron or steel work of the vessel.

"All wrought-iron and steel pipes and tubing, for salt water,

to be galvanized after they have been bent and flanges fitted.

"Care will be taken that all copper pipes are led sufficiently high to keep them out of the bilge water under ordinary circumstances.

"Feed pipes will be placed, where possible, well above the floor plates.

"Pipes will be led in such a manner that no stiffeners on bulkheads or beams on decks will have to be cut.

"Removable plates will be fitted on bulkheads where necessary to allow flange bolts to be properly set up; these plates to be fitted and made watertight to the satisfaction of the Bureau of Construction and Repair.

"Where pipes pass through watertight bulkheads the holes will be made watertight, by stuffing boxes, flanges, or other approved means, to the satisfaction of the Bureau of Construction and Repair. In no case will any of the watertight structure of the vessel form a part of the piping.

"Where pipes pass through wooden decks, they will be fitted with a hollow casting and a stuffing box, so arranged that non-conducting material may be fitted between movable and fixed parts, the details to be satisfactory to the Bureau of Construction and Repair.

"Where pipes pass through coal bunkers they will be protected by galvanized-iron casings, made in sections, easily removable for repairs.

"When such pipes are clothed or lagged the galvanized-iron casing will be dust tight.

"Pipes will not be led under openings of coal chutes.

"*Material of Pipes.*—Pipes will be of the following materials:

Pipes.	*Material.*
All pipes less than 2-inches diameter.	Copper, unless otherwise directed.
All steam pipes 2-inches diameter and above.	Steel, seamless drawn. Copper when directed by Bureau.
All feed suction pipes.......................	Copper, seamless drawn.
All feed discharge pipes.....................	Copper, seamless drawn.
Pipes to reserve feed-water tanks.......	Lap-welded wrought-iron or steel tubing, galvanized.
All steam, air and water pipes of refrigerating plant.	Copper, unless otherwise directed.

Pipes.	*Material.*
Steam fire-extinguishing pipes to bunkers, outside bunkers.	Copper.
Steam fire-extinguishing pipes to bunkers, inside bunkers.	Galvanized iron.
Cooling pipes in refrigerating room.	Galvanized iron.
Cooling pipes in scuttle butts...........	Copper, well tinned on the outside.
Suctions from bilge, crankpits, shaft alleys, and C. and R. drainage manifolds.	Lap-welded wrought-iron or steel tubing galvanized.
All internal pipes in boilers..............	Brass or steel, No. 14 B.W.G.
All water-service pipes on main or auxiliary engines.	Brass, polished above floor plates.
Copper and brass pipes 6 inches diameter and less.	Seamless drawn.
Other pipes....................................	As directed and approved.

"*Thickness of Pipes.*—All pipes will be of approved size.

"The thickness of pipes will be found by the following formulae:

"For straight copper pipe—

$$T = \frac{P \times D}{8,000} + \frac{1}{16} \text{ inch.}$$

"For steel steam pipe—

$$T = \frac{P \times D}{10,000} + \frac{1}{8} \text{ inch.}$$

Where P = pressure above atmosphere in pounds per square inch;

D = inside diameter of pipe, in inches;

T = thickness, in inches.

"For the following pipes:

Pipes.	P =
Steam, bleeder, blow pipes, suction pipes from boilers,	265, boiler pressure.
Feed discharge pipes	$1\frac{1}{4} \times 265$.
Fire main connections, copper water-service pipes, feed-suction pipes..	150.
Other water pipes without pressure..........................	50.
Exhaust pipes $4\frac{1}{2}$ inches diameter and less	20.
Exhaust pipes above $4\frac{1}{2}$ inches diameter....................	50.

"All pipes not included in the above will be of approved thickness.

"No bend will be allowed in copper pipes of which the radius is less than one and one-half times the bore of the pipe. Bends in steel pipes will be as approved.

"In determining the thickness of bent pipes the neutral axis will be considered to lie on the inner surface of the bend, and when, in bending, the outer surface is reduced in thickness more than $\frac{1}{32}$ inch, the pipe must be one gage thicker than given by formula.

"*Expansion or Slip Joints.*—Compostion.

"Expansion or slip joints of approved pattern will be fitted where required, and connecting pipes will be anchored in an approved manner. Each slip joint will consist of a composition stuffing box, gland, and entering pipe, the stuffing box and entering pipe having flanges for connecting with the pipe main, and there will be stop bolts and flanges for limiting the motion.

"All slip joints will be packed with approved metallic packing.

"All steam, exhaust and feed piping not fitted with slip joints will be run as approved, with ample bends, to provide for expansion. Such pipes will be so fitted as to put them under stress when cold by allowing for every 10 feet of length when bolting up a clearance between flanges of $\frac{3}{16}$ inch for steam pipes and $\frac{3}{32}$ inch for exhaust and feed pipes.

"*Systems of Piping.*—There will be the following systems of piping, together with such others as may be necessary to complete the machinery in accordance with the specifications and plans, viz:

"Main steam and exhaust piping; auxiliary steam and exhaust piping, with branches to all auxiliary and special steam machinery in the vessel; bleeder pipes; live-steam pipes to intermediate and low-pressure receivers; dynamo steam and exhaust piping; main-feed suction and discharge piping; auxiliary-feed suction and discharge piping; bilge suction and discharge piping; sea suction and discharge piping; fresh-water suction piping and discharge pipes to reserve-feed water tanks; discharge piping from pumps to fire mains and sanitary

system; water-service piping; boiler blow, pumping-out, and internal piping; distiller and evaporator piping; vapor, escape, drain, indicator, radiator, and fire-extinguishing piping, and steam piping to sea valves, refrigerator piping complete.

"Each system will be complete in itself, and all piping, together with valves, fittings and connections, will be shown on the working drawings."

PIPING SYSTEMS.

The following systems are met with in the Navy.

Main steam-pipes are made of seamless-drawn steel, with male and female flanges, and with gaskets of corrugated copper. The main steam line is arranged symmetrically in two systems, one on each side of the vessel. The branches from the boilers connect to the main steam pipe, which is successively enlarged till it connects to the after boilers, which size is then carried to engine room. Stop valves are fitted near the forward bulkhead of each boiler compartment for the purpose of cutting out any boiler room. Cross connections between the two systems are made generally at two joints, these are fitted with valves so that each system can be kept separate.

Inside the engine-room bulkhead there is another cross connection. A separator is fitted for each engine, also a bleeder connection to condenser, a connection to the auxiliary exhaust line, a branch for live steam to receivers and the branch leading to the throttle valve in which the main-engine stop valve is located.

Expansion joints are fitted generally between each bulkhead, and should have such a joint between each point of anchorage.

Drains are fitted to all places where water may collect, and these are usually connected to traps, which discharge to feed tanks.

Auxiliary steam pipes are of seamless-drawn steel, with male and female flanges, and gaskets of corrugated copper, except for pipes of small diameter where rubber insertion or strengthened asbestos packing is used.

In the latest practice the auxiliary steam pipe takes steam

from the nozzle at the cross connection between the main steam pipes in engine room. The pipes pass along outboard sides of engine room and form a connecting loop. A branch pipe leads aft to supply steam to the after auxiliaries, expansion joints or expansion bends are fitted at intervals, drains and separators are fitted where pockets are formed.

From the forward cross connection of main steam pipe in boiler rooms there is a branch pipe to supply steam to forward auxiliaries.

Branch Steam and Exhaust Pipes.—The general requirements for these may be seen from an extract of latest machinery specifications.

BRANCH STEAM AND EXHAUST PIPES AND VALVES.

"The size and lead of branch pipes to parts of ship outside of the machinery spaces will be subject to the approval of the Bureau of Steam Engineering. Such branches will always have stop valves in the engine or firerooms.

"1-inch branches from the auxiliary steam pipes will lead to the bottoms of main, auxiliary and dynamo condensers for cleaning the tubes by boiling.

"Connections will be made to the auxiliary steam pipe for supplying steam to sinks, steam table in pantries and crew's lavatories.

"Steam for galleys will be taken from the auxiliary steam pipe, through a separate pipe with reducing valves, steam gauge, relief valve and stop valves where approved.

"Each sea injection valve will have a steam connection of approved size for cleaning the strainer. This steam connection will be a branch pipe, with a valve at each end, leading from the auxiliary steam pipe to the injection pipe outside of the injection valve but inboard of the sea chest proper.

"Steam pipes to sea valves will be of the following sizes:

Size of sea valves.	Steam connections.
6 inches and less	¾ inch.
Above 6 inches and including 9 inches	1 inch.
Above 9 inches and including 12 inches	1¼ inches.
Above 12 inches	1½ inches.

"All branches leading from the auxiliary steam pipe to crank engines will lead from the top or side of the pipe and in no case from the bottom. When the branch leads to a lower level the stop valve on the branch will be placed as high as possible, so that with the engine standing idle there will be no opportunity for water to collect in the vertical pipe leading to it.

"All branches leading from the auxiliary steam pipe to direct-acting pumps will lead from the bottom of the pipes.

"All auxiliary machinery will take steam from the auxiliary steam pipe and exhaust into the auxiliary exhaust pipe, except as follows:

Engine.	Takes steam from.	Exhausts into the—
Main circulating pumps, fire and bilge pumps, reversing engines.	**Cross-connection pipe between the the main and auxiliary steam pipes.**	**Auxiliary exhaust pipe.**
Main air pumps..................	**Cross-connection pipe between the main and auxiliary steam pipes.**	**Do.**
Brine pump.........................	**Auxiliary steam pipe, also from evaporator tubes near discharge to traps, or from top of traps, to prevent the evaporators from becoming air bound.**	**Do.**

"All branches and sub-branches of auxiliary steam and exhaust pipes must (unless excepted) have valves at junctions and ends.

"Each auxiliary engine will have stop valves in both steam and exhaust pipes, as close to the cylinder as possible.

"When a pump or an engine, connected with the auxiliary exhaust pipe, lies below the pipe, the stop valve next the pipe will have a spindle of sufficient length to be worked from below.

"All exhaust pipes leading to the condenser from engines above the protective deck will be fitted with valves below the protective deck."

FURTHER DETAILS.

Such auxiliaries as require it have their branch steam lines fitted with reducing valves.

The former practice was to have the auxiliary steam pipe

connect to all the main stop valves, and a connection from auxiliary steam line to main steam line in each boiler room. This is the system installed on most vessels now in commission (January, 1906).

Auxiliary Exhaust Pipes are of copper, having flanges faced and grooved, steel bolts, composition nuts, copper gaskets, except for small pipes where strengthened rubber packing is used.

The auxiliary exhaust line forms a loop throughout the machinery compartments. All of the auxiliary engines have connections to it.

The auxiliary exhaust has connections so that it can be turned into either main or auxiliary condenser, into either L.P. receivers, into either feed-water heater or, into the atmosphere through the after escape pipe.

At the connections to the condensers there is a stop valve and a spring relief valve opening toward the condensers for the purpose of regulating the pressure in the exhaust line when the exhaust is turned into feed the heater or L.P. receivers. At the connection to the escape pipe two valves are generally fitted to minimize the chance of air leaks.

On older vessels, especially where no feed-water heaters are installed, spring relief valves are not fitted on the connection to the condensers.

Dynamo steam and exhaust pipes are of same material as auxiliary steam and exhaust pipes.

The latest practice is to have two dynamo rooms, one forward and one aft. The steam is then taken from the forward cross-connection of main steam line and from a connection in after boiler room. The control valves are in the dynamo rooms. Separators and reducing valves are fitted in dynamo room, and drain pipes and traps supplied. The pipes are so arranged that in case of break at least one-half of the dynamo engines in each room may be operated.

The exhaust from the dynamo engines can be directed to the dynamo condenser or to the auxiliary exhaust pipe.

In older vessels there are often fitted on boilers, separate

dynamo stop valves with a special system of piping. Separate dynamo condensers are only fitted on the latest vessels.

Steam Piping to Receivers.—This is taken from the main steam pipe and has branches leading to the I.P. and L.P. receivers. Stop valves are fitted close to main steam pipe and at each receiver, where the valve is operated by rod or lever from the starting platform.

Bleeder Pipes.—Bleeder pipes lead from the main steam pipe, forward of separator, to each main condenser. A stop valve is fitted at the connection to the main steam pipe, and there is another valve close to the condenser. One of these valves is fitted to be operated from or near the starting platform.

Escape Pipes are fitted abaft of each smoke stack and have branches leading to the safety valves of the boilers connected to their respective smoke stacks. The after escape pipe has a connection to the auxiliary exhaust line.

Vapor pipes are led from the feed and filter tanks and connect to the after escape pipe. The purpose of these pipes is to ensure an atmospheric pressure in feed tanks and to allow escape of air.

Pipes to Whistle and Siren.—These are branch pipes leading from the main or auxiliary steam pipes in the boiler room from some point that is likely to be always under steam.

Fire-Extinguishing Pipes to Bunkers.—These pipes lead from the main or auxiliary steam line to the bunkers and have valves at the bunker bulkheads. Bunkers above the protective deck sometimes have valves worked from the deck above by gearing. The pipes in the bunkers are fitted with special nozzles for distributing steam throughout the coal. In the latest practice this system is not fitted, and dependence is placed on fire hose or special water-sprinkling pipes.

Automatic System of Closing Valves.—On some late vessels the stop valves of boilers are closed by steam operating a piston connected to the valve, the system of piping for operating these valves takes steam from the main steam line. The control valves are located so that they are operated from the adjacent boiler compartment or from a station above the boiler.

This system or variations of it have in some cases been of questionable utility and reliability. Direct mechanical control is generally regarded with more favor by the operating personnel.

Fire-Extinguishers on Grates.—This system of piping is fitted on recent vessels. Nozzles are fitted for the furnaces, and the piping is either connected to the fire main or from the auxiliary pumps in adjacent boiler compartment. In some vessels these extinguishers take their water from the sea valve.

Separators.—The object of the separator is to extract the moisture contained in the stack. This is generally accomplished by an enlargement of the volume and a sharp turn around a baffle plate. Other types of separators give the steam a whirling motion, and the particles of water are thrown out by centrifugal force. In practice the simple separator gives about as great satisfaction as the more complicated ones fitted with elaborate baffling systems.

Separators must be frequently drained and the interior should be examined from time to time and have the grease deposits removed.

Expansion Joints.—To allow for expansion and contraction in piping due to change of temperature and also for movement of the structure of the ship, vibrations, etc., expansion joints are fitted. The joints are packed with metallic packing, or with strengthened woven asbestos packing. For small pipes and in feed lines bends instead of expansion joints are introduced to allow for the movement of pipes. By use of bends no danger of leakage is introduced and there is a saving of weight. Expansion joints will generally be found only on the larger piping

Packing for Expansion Joints.—This is usually metallic, reinforced by asbestos woven rings. Specially prepared asbestos woven rings are also used without metallic packing.

Lead Piping.—Feed pipes are made of copper with composition flanges, steel bolts and composition nuts, and is packed with copper gaskets, wire gauze and red lead, or with thin wire insertion packing.

The auxiliary feed suction pipes take their suction from the feed tank connecting pipe, or directly from feed tank, and lead forward to the boiler compartments. They have suction connections to all the auxiliary feed pumps in the boiler rooms.

The main-feed suction pipe takes its suction from the cross-connecting pipe of the feed tank or from the feed tank direct, and leads to the suction of main feed pump.

Feed discharge pipes lead from discharge side of main feed pump (through feed heater if fitted on suction side of pump) and then to the boiler compartment, having branches to discharge to boilers on its own side of the ship and a cross connection to the opposite side. Gate valves are fitted to cut out portions of piping not in use.

On many vessels there is only one main feed line which supplies both sides.

The auxiliary feed pumps discharge into a fore-and-aft delivery pipe having branches so that all boilers on that side of the ship can be fed.

The lead of piping differs for nearly every vessel, both main and auxiliary feed systems are, however, installed in nearly all cases.

The cut-out valves in the feed-line main are usually gate valves, to give a clear opening and reduce friction. The branches to the boilers have either stop valves or gate valves.

Feed pipes are fitted with bends to allow for expansion. Feed pipes should be covered with non-conducting material and are, as much as possible, carried above the floor plates.

Bottom and Surface Blow Pipes.—The bottom and surface blows of boilers are connected with a system of piping which connects to the overboard discharge valve in its own or adjacent compartment. This system of piping is connected to the fresh-water manifold of the auxiliary feed pump in the boiler compartment. This connection is for use in pumping out boilers.

In the latest vessels contracted for the auxiliary feed pumps have no salt-water connections whatever. This is to effectually guard against the possibility of salt entering the boiler feed.

Hotwell and Fresh-Water Pumping System.—Where hotwell pumps are fitted they have a suction to the cross-connection pipes between the feed tanks and to a cross-connection between the two air-suction pipes, to suctions on ship's side for taking on board fresh water, and to reserve tanks for the purpose of distributing fresh water. These pipes are of copper, flanges faced and grooved, and packed with approved rubber packing.

The hotwell pump has discharge piping to the reserve tanks by means of a combination manifold where one pipe can be used for either suction or discharge.

The hotwell pump discharge to feed pump suction is arranged so as to pump through or by-pass the heater, if suction heater is fitted, and then to feed suction pipe. Where pressure heaters are fitted, the hotwell pump simply discharges into the feed suction, its object being to enable the main feed pump to obtain a good suction. A great many vessels have no hotwell pump, but there is usually a fresh-water pump for handling fresh water in reserve tanks and for pumping fresh water on board.

The piping to and from the reserve-feed tanks in the latest vessels is of wrought iron or steel, galvanized, the same as bilge and drainage piping.

Air-Pump Piping.—This consists of (1) a suction pipe from condenser to air pump. (2) A discharge from air pump to feed and filter tank, and a connection running across the ship, connecting the two air-pump suction pipes. By means of this cross-connection either air pump can work on either condenser. This cross pipe also has connections to the hotwell pump or to some of the feed pumps, so that these pumps can pump direct from the condenser. Such a connection would enable the condenser to be used although the air pump should be broken down.

The exact arrangement of air-pump piping can be understood only by an examination of the piping plans for a vessel.

Salt-Water Suction and Discharge Piping.—Pipes for circulating water for main, auxiliary and dynamo condensers.

These pipes are of copper, have composition flanges packed with rubber packing.

Zinc protectors are fitted in standard zinc boxes for preventing corrosion of the pipes. Joints in such piping are made in accordance with Standard No. 86, issued by the Navy Department.

DRAINAGE SYSTEMS.

Drainage Piping.—The drainage piping of large vessels consists of a main and a secondary drain with their suctions, a double-bottom pumping system for the inner-bottom compartments other than the reserve feed tanks, the connections from these systems to the pumps, and the discharge pipes from the pumps overboard.

Main Drain.—The main drain is a pipe made of steel; for large vessels it is about fifteen inches in diameter. This drain extends throughout the machinery spaces and has a suction in each boiler and engine compartment. Each suction valve is fitted with gear for operating from above the protective deck.

The main drain has pumping connections to the bilge pumps in the engine rooms and to the auxiliary feed pumps in the fire rooms, on all but latest vessels, those contracted for after 1904. These connections are either direct or through the drainage manifold, which in late vessels is separate from the engineering manifolds at pumps, having connections to main drain, secondary drain and double bottoms.

The main drain also has connections to each main circulating pump in the engine room.

On the latest vessels the connections from the drain to fireroom, fire and bilge pumps have been omitted.

Secondary Drain.—This is a pipe varying in diameter from five to seven inches, and is located along the center line of the ship on the side opposite from the main drain. This drain is connected, in forward boiler room, to a manifold which has valves controlling the suctions to the forward parts of the vessel outside of the engine-room compartments. In the after part of the engine room it connects to a similar manifold which

controls the suctions to the after compartments of the vessel. The secondary drain has suctions in each engine and boiler compartment with valves worked from above the floor plates, and on many vessels arranged to be worked from deck as well.

The secondary drain has pumping connections to the fire and bilge pumps and to the auxiliary feed pumps. These connections are either direct to the pumps or through the drainage manifolds. In the latest vessels contracted for after 1904 auxiliary feed pumps are not so connected.

Independent Bilge Suctions.—The bilge pumps in engine rooms have an independent bilge suction pipe, which has suctions to engine bilge, crank shaft, and, in latest practice, to shaft alley. Additional branch suctions are sometimes fitted to this pipe for the purpose of pumping out pockets difficult to keep dry.

The auxiliary feed pumps or boiler-room fire and bilge pumps have an independent bilge suction to the bilges of their own boiler compartments.

Strainers are fitted in all suction pipes for the purpose of collecting dirt from the bilge and preventing it from clogging pipes and pumps.

Valves and Manifolds.—The valves in manifolds at pumps are usually check valves arranged so that they can be raised off their seats by an extreme movement of the valve spindle, usually called stop lift check valves.

In order to pump a compartment through the secondary drain it is necessary to open the suction valve for the compartment, the valve drainage manifold which controls secondary drain suction, and the stop valve separating drainage manifold from the pump manifold.

Double-Bottom Piping.—The double bottoms are connected to a double-bottom main, which is arranged so that the fire and bilge pumps or auxiliary feed pumps can pump on them either by a direct connection to the pump manifold or through the drainage manifold. Valves controlling the suctions to the

compartments are located above each compartment and are arranged to be operated from above the floor plates.

On other vessels the system is arranged so that independent pipes, one from each compartment, lead to double bottom manifolds located in boiler and engine rooms.

The system of double-bottom piping with its manifolds and valves is under the cognizance of the Bureau of Construction and Repair.

Water-Service Piping.—The water-service piping consists of brass-piping for conveying the water to various engine and thrust bearings.

On some vessels the water service takes its water from the discharge of main circulating pump, and the water after leaving the bearing or guide, instead of going to the bilge is allowed to discharge to main circulating-pump suction, thus avoiding a very large amount of water in the bilges and crank pits.

Flushing System.—Material of pipe, copper. This system consists of a main supplied by a connection from the distiller circulating pump and also usually by a connection from the fire main. This pipe has branches supplying water to water closets, urinals, wash rooms, shower baths, laundry, pantry and galleys. Relief valves are fitted to prevent an excessive pressure coming upon the pipe.

This pipe, from a flange located near the distillers, is under the cognizance of the Bureau of Construction and Repair.

Fire Main.—Copper.—This system, on large vessels, consists of two large mains, one on each side of vessel, directly underneath the protective deck within the machinery spaces. These two systems are cross connected and are connected by pump risers with the fire and bilge pumps in engine rooms, the fire and bilge or auxiliary feed pumps in boiler rooms and, in some vessels, with the distiller circulating pumps.

The risers from the pump are fitted with cut-out valves near the main. From the main on each side risers are taken, usually through some of the machinery hatches, and have plugs and short branches on the various decks.

Each riser is fitted with a cut-out valve below the protective deck close to the main. These valves are arranged to be worked from above the protective decks. By this arrangement it is possible to cut out any riser when damaged, without impairing the rest of the fire main.

The fire main has connections for flooding the magazines above the lower platform. There are also connections to each ash chute for flushing same. These last should more properly be connected with flushing system.

Plugs are arranged so that any part of vessel can be reached with a fifty-foot hose from two plugs.

All valves in fire main are composition gate valves. Chapman valves are largely used. The relief valves on fire main are set at one hundred pounds pressure.

Zinc boxes are fitted in the fire main to prevent corrosion.

The principal difficulty experienced with fire mains is the corrosion of pipes due to the action of salt water, and leaky valves. Of late most gate valves are fitted with removable seats which can be taken out and faced off in a lathe when the seat becomes rough.

All fire-main plugs are either 2½-inch or 1½-inch Navy standard.

The fire main and all risers from it are under the cognizance of the Bureau of Construction and Repair. The risers from the steam pumps to the fire main are under the cognizance of the Bureau of Steam Engineering.

The division of cognizance between the two Bureaus is at the flange on the main between the riser and the main.

REMARKS ON ERECTING PIPING ON BOARD SHIP.

Steam Piping.—A great deal of the trouble experienced with steam piping is due to want of alignment. This want of alignment operates to throw excessive strains on the flanges of stop valves, anchorages, separators and expansion joints. These strains are brought about by reason of the flanges being brought together by forcing on the connecting bolts instead of the joints coming fair without this outside assistance. Any

joint that requires outside pressure to make the parts come together will cause trouble. When the flanges do not come together fairly they should be taken down and refaced, and, if necessary, a thin steel distance piece put in to make the required length. When erecting heavy pipes, every length should be placed in position and properly supported and aligned in its own brackets and hangers, before being permanently secured. It will usually be found that various lengths will have to be altered slightly before proper alignment is obtained.

Expansion and contraction due to change in temperature, to vibration and to slight movement of the structure of the vessel must be properly allowed for. To this cause most leakage in joints is due. The positions of anchorages, bends, valves and branch connections will materially affect the ability of the piping to accommodate itself to the changes.

When certain joints are continually leaking, the connections should be carefully studied; often some slight alteration can be made in the anchorage, connections, hangers or lead of pipe that will allow for the necessary movement and prevent the strain which causes the leak coming on the joint.

Vibration is a prolific cause of leakage, especially in small steam piping. Vibration of pipes can often be largely stopped by placing bands or guys on the pipes that will stop the vibrating movement but which will not prevent expansion or contraction due to temperature or to regular movement of the vessel. Often the pipe can be prevented from vibrating by taking a turn of wire around it and securing the wire to some part of the ship's structure.

Position for Drainage.—Steam pipes should be erected so that they will naturally drain themselves and so that pockets, where condensed water may accumulate, may be avoided. The danger from the accumulation of water is two-fold: (1) water ram; (2) injury to packing. The pipe lines should be run so that the water of condensation will drain either back to the boilers or to the separator. Where the piping is long and complicated, drains should be fitted at frequent intervals.

Pockets where water collects are often formed by stop valves

being placed so that their bodies form a water trap. An example is shown in cuts (Lunkenheimer Co.'s catalogue) Fig. 1. In the first figure the arrangement permits the accu-

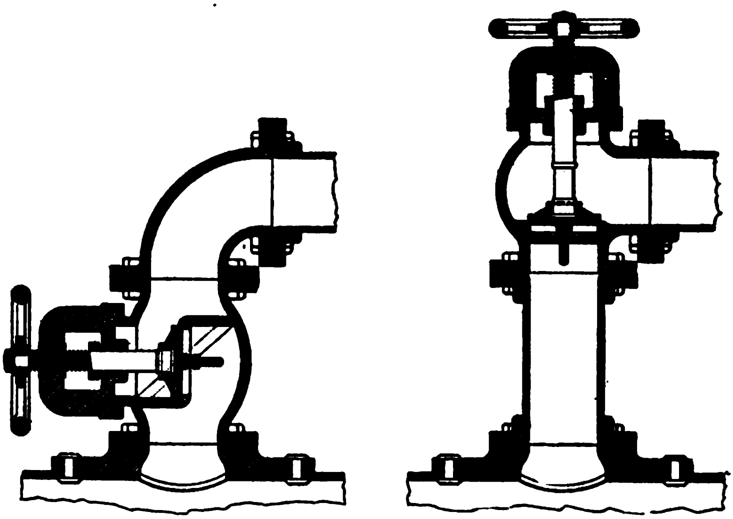


Figure 1.

mulation of condensed water above the valve when it is closed, and by careless opening water hammer may result. In the second figure the forming of a water pocket is avoided. In any case where such pockets are unavoidable a drain should be fitted, and this drain should be open before the large valve is opened.

Care in Fitting.—The greatest care should be exercised in fitting pipes to see that all dirt, grease, parts of gaskets, scale or little pieces of metal are not left in the piping; since this foreign matter is liable to get into the valves and cause them to leak. In smearing flanges with grease care should be taken that this does not get into the piping and valve castings, since it will be carried to the bearing parts of the valve and there, owing to its sticky nature, it will hold grit and scale on the seats and cause them to become scored.

Corrosion of Piping.—Internal pitting, due to air, is liable to be met with, and the interior of pipes should occasionally

be examined to see if any is present. External corrosion is also liable to result from the combined action of the heat and moisture on asbestos pipe covering, causing pitting. This outside corrosion can be guarded against by giving the pipe a good coat of graphite paint before the covering is put on.

Repairing Leaks.—Leaks in copper piping are permanently repaired by rebrazing the seam or by brazing on a patch. To braze on a patch, clean the metal carefully with file, emery cloth and hydrochloric acid. Cut out a piece of copper of suitable thickness and shape it carefully to the pipe, cleaning thoroughly the surface next to the pipe. The patching piece is securely wired in place. The pipe is then heated in a forge with a charcoal or coke fire. Spelter or hard solder, with borax as a flux, is placed over the hole on the inside and the whole heated, the pipe is moved and turned about to keep the brazing matter from running away from the patch. As the brazing mixture melts, it runs betwen patch and surface of pipes and makes a joint between the two. If the hole is very small, the brazing mixture may have to be put on the outside and caused to run in from one side of the patch. Great care must be exercised to prevent the pipe from burning. By the use of loam the heat may be directed to any one position and protect the other parts from injury.

A patch may be welded, riveted or bolted on a steel pipe. If the leak is a small one on a thick pipe, the hole may be rounded out and a rivet put in from the inside and hammered over from the outside. Or the hole may be tapped out and a small plug screwed in and riveted over on the outside. It is, however, very difficult to make a permanent repair with the pipe in place, though often there is not sufficient time to take pipes down, and a temporary measure must be adopted.

Some Methods of Temporary Repair.—Place a putty made of red lead and oil, sometimes mixed with pieces of hemp to give strength, over the leak, wrap with canvas, and then serve the pipe with marline. Or place a piece of packing over the putty and then serve with wire. Another way is to put on the putty and then serve the pipe with wire, and solder the wire

on as it is being served. Clamps made of iron or brass and drawn up with bolts are often useful in holding the patch over the leak. A very secure patch can be made by shaping a piece of copper to the leak, then put a piece of soft packing coated with red lead between the pipe and copper and draw the piece of copper up with several bolted clamps. Pine plugs driven into holes form a ready means of stopping a leak in low-pressure water piping. After the plug is driven, the moisture will cause it to swell and continue to make it tighter. Pin holes may be closed by the use of soft solder. Portland and smooth-on cements are also very useful in repairing leaks. Smooth-on cement is especially good for cracks in metal piping or casting.

Brass and copper piping in bilges should be kept well covered with paint. If there are bare places and the bilge water is brought into contact, galvanic action is liable to be set up.

Copper water piping is peculiarly liable to electric corrosive action. The use of zincs acts in some manner as a preventative, but where there is a continual flow of salt water it appears to be impossible to prevent the corrosive action. Tinning the pipes also acts as a preventative to some extent, but it does not stop the action. In order that the various sections of piping may not be insulated from each other by rubber gaskets at flanges, the flanges can be connected by short pieces of copper.

DRAINS AND TRAPS.

Places where condensed steam can accumulate are provided with drain pipes with or without traps for conducting such water to the feed tanks or condensers.

The parts usually fitted with drains are as in the accompanying table.

MACHINERY DRAINS.

Parts drained.	Drains through—	Drains to—
Main steam separators....	Approved automatic "float trap" with by-pass.	Main condenser, feed tank near bottom.
Dynamo steam separators.	Do.	Dynamo condenser.
Exhaust to low-pressure receiver (separator if directed).	Do.	Main condenser.

Parts drained.	Drains through—	Drains to—
Dynamo exhaust pipes (separator if directed).	Trap with by-pass and pumps (if directed).	Exhaust to dyuamo condenser, above valve at condenser.
Aux. steam separator, steam pipes and valves.	Approved automatic "float trap" with by-pass.	Main condenser, aux. con., feed tank near bottom.
Whistle pipe and separator.	Do.	Feed tank near bottom.
Low-pressure jacket drain (¾-inch diameter).	Approved automatic "float trap" with blow through, or "expansion trap."	Lower part of feed tank.
High-pres. evap. coils.....	Do.	Feed tank, main and aux. cond.
Low-pres. evap. coils......	Do.	Same, with branch to dis. fresh-water pump suction.
Heater drains (each man.)	Do.	Feed tank, main and aux. cond.
Galley steam drains.......	Do.	Feed tank, aux. cond.
Cylinder drains, main engine.	No trap, unite in pipe.	Top main cond. and to bilge.
Cylinder drains, aux. eng.	No trap......................	As approved.
Cylind. drains, dyn. eng.	No trap......................	Dyn. cond. above valve.
Cylinder relief valves.....	No trap. Drain pipes.	Bilge.
Valve-chest drains..........	Do.	Main cond. and bilge.
Feed-water heaters.........	Trap if required..........	Feed tank, main and aux. cond.
Where hot water passes...	Float traps *must* be used.	As approved.
Where warm water passes.	Float or expansion traps.	Do.
Bath heaters......	No trap......................	Most convenient aux. exhaust in continuous operation.

Drain pipes are of copper, and are fitted with unions where necessary so that the pipes can be readily disconnected for overhauling.

Steam traps are now fitted of many different kinds. They are of two general types: (1) the float trap, (2) the expansion trap.

The following extracts from an article by Lieutenant R. T. Winston, U. S. Navy, show the results of recent experience with steam traps in the U. S. Navy.

STEAM TRAPS.

"*Types of Traps.*—The open-bucket float type, though bulky and somewhat affected by the ship's motion in a seaway, has given the most general satisfaction. It will take care of a sudden gush of water and requires no adjustment for varying pressures and temperatures.

"The hollow-ball float type has given some trouble on account of the ball corroding through, getting punctured or becoming detached. This type should give satisfaction if proper provision has been made to insure watertightness of the float and if the float is securely attached to its lever.

"The expansion type of trap is superior to the float type in that it is not bulky, has few parts and lasts well. It does not work well where there are wide variations in the temperature or in the amount of water to be handled. Under such conditions trouble has been experienced with the frequent adjustments, irregular working of the trap and the leakage of steam.

"The following traps are largely used in the Navy: (float type) Dinkel, Nason, McKellar and Kieley; (expansion type) Geipel.

"*Requirements of a Good Trap.*—The best trap is the one that will discharge the water which collects without wasting steam and at the same time requires little attention. Other qualities to be considered are durability, simplicity, facility of overhauling, weight, size and cost.

"It is better to have the drain valve of a trap at the top of the chamber rather than at the bottom, where sediment collects. It is also an advantage to have valves outside the float chamber, where they may be cleaned and overhauled without breaking a joint.

"The bonnet of a float trap should be secured with studs, or if through bolts are used, the heads should be so fitted as to prevent turning. The facility for overhauling might be still further increased by the use of hinged bolts.

"The piping for all traps should be fitted with union couplings to facilitate the removal of trap for examination or re-

pair. In case of an expansion trap, the discharge should be fitted with a by-pass to the bilge, by means of which it could be determined whether the trap was working properly.

"*Traps for Special Use.*—Where the quantity of water is subject to sudden variations, as in steam pipe and separator drains, the float type should be used. Where there is a slow, regular and continuous condensation at a practically constant temperature, the expansion type is satisfactory and often desirable.

"For cylinder jackets of main engines an arrangement similar to that on vessels of recent designs seems best; *i. e.*, steam goes from the H.P. jacket to the I.P. jacket through a reducing valve from the I.P. jacket to the L.P. jacket, through another reducing valve, and then drains through the I.P. jacket trap; one trap taking care of the water for the whole system.

"The following easily improvised substitute for a jacket trap has been used with success: A small cast cylinder, fitted with a drain valve and a gauge glass, is placed in the jacket drain-pipe line. When in operation, the drain valve is opened just enough to keep the water in gauge glass at a fixed height, the whole arrangement being near the starting platform, and readily seen.

"For steam radiators the bucket type generally gives the best results. Expansion traps when used should be placed close to the radiators, and each trap should not drain more than two or three coils; they have been found to give considerable trouble when required to drain more than one heating circuit.

"For evaporators and feed heaters bucket traps have been extensively used. It is probable that an expansion trap, carefully installed, would work efficiently on evaporator coils.

"Drains for feed-water heaters when discharging to condensers do not require traps, but if the heater drains discharge to the feed tank traps may be necessary.

"For steam separators and steam lines bucket traps must be used.

"*Location of Traps.*—A trap must be located below the lowest part of the vessel or pipe to be drained, and, if practicable, above the level of the tank into which it discharges, in order to prevent water backing up when the trap is not in operation. This applies particularly in the case of the expansion trap, which has given trouble in discharging against a head of water. Under certain conditions it may be necessary to fit a non-return valve in the discharge line.

"All traps, and particularly those which will need frequent overhauling, should be so placed as to be easily overhauled and examined. If a trap cannot be installed where it is fairly accessible for overhauling it should be omitted and a drain valve fitted instead.

"*Operation and Overhauling.*—Many traps are of no value on account of lack of care and attention.

"A float trap can be heard 'dumping' when in order, and may work satisfactorily for months without overhauling. Whenever the sound of dumping is not heard at intervals the trap should be examined at once. Less trouble is experienced with a large trap than a small one, as in the latter case valves are more easily clogged, thus causing a failure to discharge, or if not entirely stopped will permit a leakage of steam. This is particularly the case where there is likely to be sediment in the water.

"While experience is necessary to judge in each particular case the period a trap may be left without examination, it may be said in general that a large trap should never go more than three months or a small trap more than two months without examination and cleaning.

"By opening the expansion-trap discharge by-pass to bilge it may be seen whether the trap discharges water properly, and if it leaks steam. The trap should be tested frequently when in use and adjusted and cleaned out if necessary."

Draining Steam Lines.—Steam lines and separators are fitted with drains usually led to traps, the discharge from the trap being led to the feed tank or the auxiliary condenser.

On some vessels main separators are also drained by leading

a pipe from the bottom of the separator for operating some reciprocating pump, such as a hot-well pump. Dynamo separators are sometimes drained by leading a pipe from bottom for operating dynamo condenser pumps. By this system the separators are being drained continuously.

Draining Jackets.—In order that steam jackets may work efficiently they must be kept constantly drained. If jackets fill with water they do not impart any heat to the steam in the cylinder.

In the Navy there are two different systems for draining jackets. On late vessels, steam first enters the H.P. jacket, the discharge leads through a reducing valve to the I.P. jacket, then through another reducing valve to the L.P. jacket. The discharge from the L.P. jacket leads to a trap. The reducing valves placed between the jackets can be set to regulate the pressure in each of the jackets. Sometimes additional drain cocks are fitted at the bottom of each cylinder jacket to allow any water condensed in the jacket to be drawn off.

The other system is to have each jacket have its own independent steam connection, reducing valve and trap.

While jackets are being used the traps should be occasionally by-passed to ensure that water is drawn off.

Cylinder Drains.—The cylinders and valve chest of the main engines are fitted with drain cocks at the bottom. These cocks are arranged so as to be operated by shafts and levers from the working platform. One lever being arranged to move all the cocks of each cylinder and its valve chest.

The drain cocks are high-pressure packed cocks.

The discharge from cylinder drains leads to a pipe which leads to condenser, and is usually fitted with branch to bilge.

The drains are led to the condenser to ensure thorough draining. To avoid blowing through to the condenser it is most important to see that the drain cocks are tight.

Particular attention must also be paid to see that the valves in branches to the bilge are absolutely tight.

Cylinder drains of auxiliary engines are either led to traps

or to bilge. The large auxiliaries, such as dynamo engines and circulating pumps, usually have their drains led to traps. For the purpose of saving fresh water it would seem advisable to lead the drains of the various auxiliaries now going to the bilge, to the auxiliray exhaust. This has been done on some vessels.

Heater Drains.—Heater drains are led to traps which discharge to the feed tank or to the auxiliary condenser. In order to ensure pipes being thoroughly drained these traps should be occasionally by-passed. If heaters are not properly drained they fill up with water and there is no proper heating effect. Drains from the galley and pantries are led to traps.

Bath-heater drains are led to the auxiliary exhaust line. At first these drains were led to traps, but as there was not a steady drain the heaters did not work well. When connected to auxiliary exhaust there is always a thorough draining.

The principal repairs to drains consist in tightening up on leaky joints, nipples and unions, repairing split pipes, overhauling traps and grinding in and packing valves.

The principal cause for leaks is the accumulation of water in the pipes.

Valves are generally caused to leak by having their seats scored from dirt or scale in the piping, or by the valve disc and seat corroding from the action of salt, acid or alkali in the drain water.

BLOWERS AND BLOWER ENGINES.

Blowers are installed on board ship for purposes of furnishing forced draft for the boiler plant and for ventilation.

The system of forced draft almost exclusively installed, is the closed-fireroom system. On some small vessels and on the armored cruisers *California* and *South Dakota* the closed ash-pit system is installed.

The blowers are installed in the firerooms, generally suspended from the protective deck or they are located on the protective or splinter deck above the boiler rooms, where they have a more substantial support and are not so liable to shock and vibration.

When they are above the boiler rooms the engines can be much easier kept clean, they are more accessible for overhauling and attendance, and they are not required to work in such a high temperature. They can also be repaired without interfering with the operation of boilers.

Two blowers are usually fitted for each boiler compartment, so that if one should break down the other will be available. In order that the pressure may not be lost through the opening to the disabled blower, air-tight doors or shutters are fitted on the blower ducts.

Types of fans adaptable for naval service are as follows.—

Peripheral-Discharge, Straight-blade, Centrifugal Fan.— This is the type most largely encountered, and is specifically suited to the conditions for producing forced draft on naval vessels. The hub is of cast iron and fitted to the steel shaft. It has steel T arms rigidly cast into place; in other types the hub and T arms are cast in one. The steel plates are riveted to the arms, and the plates, in turn, are bolted to the conical side plates

Curved-Blade Peripheral-Discharge Fan.—For heating and ventilating fans the blades are sometimes turned backward to secure an easier escape of the air and to reduce the noise of the fan.

Cone Ventilating Wheel.—This type secures very great efficiency and is especially suited to places where air is drawn from one compartment without ducts to an adjoining one. It is usually installed without a casing. At the center of the wheel, forming its hub, is a conoidal casting with its apex toward the inlet side of the wheel, so that the entering air is gradually deflected in its course. By this means all unnecessary friction and loss of heat, due to change of direction, are avoided.

Attached to the circumference of the casting is the steel back plate of the wheel which carries the numerous curved blades. This back plate is stiffened by T steel arms radiating from an auxiliary hub. This type is used on torpedo boats for forced draft.

Disc Wheels.—For moving large volumes of air under low

pressure with a minimum expenditure of power, disc wheels are very efficient. They are especially applicable to exhaust ventilation. The fan consists of a hub to which are cast or otherwise secured radial arms to which the steel plate blades are attached. The blades, acting as inclined planes, force the air through and beyond in a direction parallel to the shaft.

Fans having their blades curved to a regular pitch, like a propeller, are also used. An increase in the number of blades does not add to the capacity of the fan, but secures smoother working.

The Sirocco Blower.—This type of fan is, like the others, based on centrifugal action, but in the blower the blades are arranged in a different way. The blades are long and narrow, and curved forward in the direction of rotation and mounted parallel to the shaft. The length of blades is about $\frac{3}{5}$ of the diameter, and the height, radially, is about $\frac{1}{16}$ the diameter of the fan. The blades are set closely together and are riveted to an iron ring at the back, which is attached to a cast-iron conical hub, the front ends of blades being riveted to an iron ring. The suction is at the center and discharge at periphery, as on other centrifugal types.

The Sirocco blower is a late development, and an increase in efficiency of about ten per cent. over the ordinary type is claimed for it.

The ventilating blowers on board naval vessels are now operated entirely by electric motors and require the usual care of such electric apparatus. The fans are usually peripheral discharge wheels with curved blades.

Forced-Draft Blower Engines.—Forced-draft blowers are operated by fast moving steam engines. They are usually two-cylinder simple or two-cylinder compound engines. On torpedo boats, yachts and small vessels horizontal engines are used. Turbine engines may also be used. The speed of these reciprocating engines is 250 to 700 revolutions per minute. On the latest vessels forced-draft blowers are to be electrically driven.

Blowers and blower engines are usually of special make. Forbes, Sturtevant, and Buffalo Forge Co. are makes very generally used.

One of the most important matters in installing blower engines is to secure sufficiently rigid supports to check vibrations at high speed. Supports to suspended engines are usually too light, and they are not sufficiently rigid. These faults can often be remedied by installing additional braces.

Operation.—The principal thing to look out for is to keep a proper supply of oil. Blowers are usually fitted with closed automatic sight-feed oil boxes, and some have forced lubrication, similar to that of the latest dynamo engines. To maintain a certain air pressure the speed necessary will depend in a large measure on the tightness of the forced-draft doors. If there is much leakage the blower engines must be run faster.

The speed at which the blower is to be run is regulated by the water tender in charge of the fireroom, who is governed by the air pressure that he wishes to maintain. On vessels where blowers are above the protective deck the admission of steam is often controlled from the fireroom by means of an extension stem from the blower branch steam pipe.

Care and Overhauling.—Blower engines require periodically a general adjustment of working parts and an examination of cylinders and valves. With well designed and well balanced engines little trouble is met with.

One of the principal troubles with blower engines is dirt and grit, dust, etc., getting into some of the bearings. Special care should be taken to see that the doors of casings, oil service, etc., is absolutely dust tight. The space surrounding the blowers should be kept clear of dust especially at the times that the engines are being overhauled. Additional dust guards can often be fitted over bearings and oil service.

In order to insure blowers being ready they should be tried at least once a month under steam. Forced-draft doors, etc., must be frequently overhauled, as they often warp out of shape from heat.

ASH HOISTS.

Ashes are usually disposed of by hoisting them to the main or gun deck and dumping them in the ash chute.

Fireroom ventilators are used for the hoists, and rails are fitted for the purpose of guiding the ash buckets. The engines are usually located in the fireroom hatch at the level of the gun deck. One engine is employed for a hoist at each side. The engine is put in gear with either side by means of a shifting clutch. A general type of ash hoisting engine is that shown in Fig. 2.

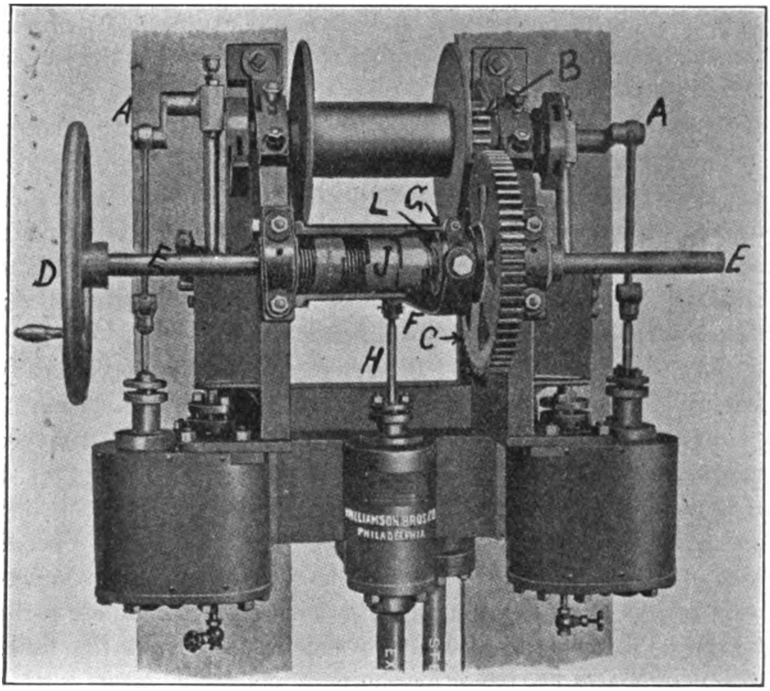


Fig. 2.

Operation.—Steam should be turned on slowly and cylinders drained. The control wheel should be turned uniformly and not by jerks. Control wheel should not be suddenly reversed. Great care is to be taken that the change clutch is properly in gear, and that it is in gear on only one side.

The detail method of operation is as follows: AA, are single eccentric cranks, which operate the main valve on each cylinder. B. is a gear wheel fixed to the drum and gearing into a spur wheel C, cast on a sleeve, L, with an internal thread to correspond to the external thread on shaft E. A groove is cut in the sleeve fitted with a loose collar, G, which, through a bell-crank lever, F, moves the reversing valve by means of the valve stem H, when spur wheel C moves to the right or left.

Turning the starting wheel, D, to the right, the spur wheel, C, moves to the left, the differential valve moves downward and the engine turns, moving the drum shaft to the left; the spur wheel, C, will then have a right-handed motion, pushing the sleeve to the right, and closing the differential valve, with the result that if the hand wheel is stopped the engine stops, or if the hand wheel is turned in the opposite direction a reverse movement is given.

The thread shown on the shaft carries a nut, J, which is prevented from turning, but has a certain lateral motion, between the safety stops, to give a proper amount of hoist. On bringing up on the stops the engine is stopped. This prevents overwinding and the breaking of cable, and stops the ash bucket as it reaches the fireroom floor. The stops can be adjusted to the desired hoist.

To Overhaul.—It often happens that parts of these engines are out of line. In some cases the stuffing boxes when packed tight may throw valve stems to one side. Parts of clutches are often made of cast iron, and sometimes break from jar. When made of composition they will stand wear and jar much better.

The wire hoisting cable should be kept clear of paint and occasionally wiped over with graphite and tallow.

Cylinders and valve chests should be periodically overhauled and coated with graphite. When drains of ash hoists lead to open air special care should be taken to see that they are closed while engine is standing idle.

Electric Ash Hoist.—This type of ash hoist has been used in some foreign vessels. The electric hoist is heavier and more

costly than the steam engine, and there are other disadvantages to cause its non-adoption for general service.

Ash hoists are located near boiler rooms so that steam leads are always short and direct.

The lift ash hoist, consisting of a long steam cylinder whose piston operates a multiple pulley, is used on some vessels, and these are often met with in the merchant service. The noise encountered with the usual type of reciprocating engine is avoided and the space occupied is very small.

Ash Ejectors.—Some naval vessels, especially torpedo boats, destroyers, and scout cruisers, which are likely to be under heavy forced draft for long periods, are fitted with ash ejectors for removing ashes from the firerooms. A connecting pipe leads through side of the ship several feet above the water line. The connection at the bottom of the hopper is led to the discharge of a pump.

The operation is as follows: The lid of the hopper is closed and the jet cock and drain cock are closed. First drain the water from the discharge pipe by opening the drain cock, and start pump. When a good pressure has been obtained, open the jet cock quickly. The water issues from the jet nozzle at a high velocity and discharges at the upper end of the pipe. It creates a suction in the hopper, and if this is now opened and the ashes shoveled in they will be sucked in and then carried up with the water and ejected from the ship's side. When all ashes are in, the hopper is secured and pump stopped.

The base of hoppers is fitted with a grating to prevent large lumps coming in and clogging the pipe.

Difficulties with ejectors arise from opening the hopper before a proper jet suction is secured, or by stopping or slowing the pump before closing the hopper. If this is done the water will back into the fireroom.

The pump for operating must be sufficiently large to maintain a constant pressure at the jet. Duplex pumps are considered more desirable for this purpose, but simplex pumps are also used. The connection from the pump should be short and direct, and the pump should be in the same compartment

as the ejector. Where ejectors are less than 4-inch diameter, considerable trouble is experienced by pipes clogging. The bent part of pipe is subject to considerable wear from the suction of ashes and clinkers. This is often made up with a specially thick cast-iron piece that can be renewed when worn.

TELEGRAPHS AND REVOLUTION INDICATORS.

Mechanical telegraphs for the purpose of transmitting orders for operating the engines from bridge to engine room are now all made to conform to a certain standard.

The principal attention that these devices requires is occasional oiling of pulleys and adjustment of the length of chain. In warm weather, or where the leads are in proximity to boilers, the heat will cause the chains to lengthen. This is especially so on vessels having a long lead.

This slack is taken up by turnbuckles placed at intervals. The spiral springs which control the bell clapper sometimes become weak and have to be renewed.

Mechanical telegraphs for transmitting orders from engine room to boiler rooms are also fitted. These are arranged so that all the dials are operated when any one of them is moved.

An electric revolution regulator to signal to the engine room to go from one to four revolutions slower or faster is also fitted. There is a light for each number on the dial, and an electric bell which rings when the contact lever is moved.

Electric speed indicators are fitted on some vessels which show the number of revolutions at which the engine is moving.

Electric Revolution Tell Tales are fitted for each shaft. A contact maker operated by a cam on the shaft, and which makes one contact at every revolution, is located in the shaft alley. This is connected with a case which has mounted on it two small index hands which move at every contact. The circuit is wired so that the ahead hand moves when engine is moving in the go-ahead direction, and the backing hand moves when backing. This is usually accomplished by a cut-out on the reversing shaft.

A mechanical revolution indicator is fitted in each engine

room, placed so as to be readily seen from the working platform. These are worked from the engines by positive movements. The dials are graduated to 100 revolutions, each graduation indicating one revolution of the engines.

The object of the above instrument is to actually demonstrate at a glance the revolutions of both the starboard and port engines, the red hand indicating the speed of the port and the green hand that of the starboard.

Provision is made for stopping either pointer, so that the engines may be quickly regulated with the aid of the indicator.

GENERAL WORKSHOP.

The general workshop on large vessels contains the following machine tools:

Large lathe;
Small tool-room lathe;
Shaper;
Milling machine;
Drill press;
Sensitive drill.

On the smaller vessels the milling machine and sensitive drill are omitted.

The following are the specifications for the machine tools for the latest battleships.

MACHINE TOOLS.

There will be provided and installed in the general workshop the following machine tools, viz:

One screw-cutting, back geared, extension gap lathe, to swing at least 28 inches over the upper and 48 inches over the lower ways, with not less than ten feet between the centers, when extended.

To have hollow spindle, compound rest, steady and follow rests, power cross feed, taper attachment, countershaft, 2 face plates, 1 large and one small chuck, one drill chuck and a complete set of at least 16 lathe tools and 7 lathe dogs.

One screw-cutting, back-geared tool-room lathe to swing

at least 14 inches over the ways and not less than 4 feet between the centers.

To have quick-change feed gear, power cross feed, hollow spindle, taper attachment, oil can, one large and one small face plate, counter shaft, one large and one small chuck and drill chuck, drawing-in attachment, with complete set of chucks, a set of at least 12 lathe tools and 6 lathe dogs.

One column tool-room shaper of at least 15-inch stroke and not less than 15-inch traverse; to have automatic stop, adjustable table, and graduated swivel vise with jaws for taper work, 24-inch graduated index center and a set of at least 12 shaper tools.

One upright drill, to drill up to $1\frac{1}{2}$ inches, to have at least 14 inches from edge of work, to have not less than 14 inches of traverse of spindle, to be fitted for No. 4 Morse taper, with requisite number of sockets.

To have a complete set of twist drills from $\frac{1}{4}$ to $\frac{5}{8}$ inch by 32ds, and from $\frac{11}{16}$ to $1\frac{1}{2}$ inches by 16ths. To have automatic and hand feed, sliding head, counterbalanced spindle, circular table, revolvable by gear and adjustable vertically by gear, automatic stop, tapping attachment, countershaft, and a small drill chuck fitted.

One 16-inch sensitive drill, to have a counterbalanced spindle fitted for No. 1 Morse taper, sliding head; table to have vertical adjustable movement, to drill holes up to $\frac{5}{8}$ inch, with drill chuck and complete set of drills from $\frac{1}{16}$ to $\frac{5}{8}$ inch by 32ds.

One universal milling machine with overhanging arm; to have at least 18-inch longitudinal feed of table, 13-inch vertical movement, and $4\frac{1}{2}$-inch traverse. Spindle to be fitted for No. 9 B. & S. taper. To have automatic cross feed, universal chuck, arbor, index, swivel, vise, spiral cutting attachment, countershaft, 3 milling cutters, 2 side millers, 6 metal slit saws, 3 angular cutters, 4 end mills and 2 collets.

One combined hand punch and shears with 6-inch shear blades, to cut $\frac{3}{4}$-inch round iron, shear $\frac{3}{8}$-inch steel plate, and punch $\frac{3}{8}$-inch holes in $\frac{3}{8}$-inch mild-steel plates 4 inches from edge.

One double emery grinder on column, with carborundum wheels, 12-inch diameter by 2-inch face, with countershaft and attachment for surface grinding and with 2 spare wheels.

One portable cylinder-boring machine, with latest attachments for boring cylinders in place. To bore from 6 to 24 inches diameter and to have at least 30 inches travel.

Six machinist's swivel bottom vises, with jaws at least 5¼ inches wide, opening 8 inches, with pipe vise jaws and copper vise lips.

One approved steel blacksmith's forge, of the portable folding type, with all the necessary tools and fixtures. The blast wheel to be about 12 inches diameter, turned by a crank. The gearing enclosed in a dust-proof case. When set up, the pan to be about 24 inches square. The pan and hood to form a case which, when the forge is dismantled and the parts placed inside, will be a strong, impact, steel box, provided with handles for carrying. This forge will be supplied with a set of tools of assorted sizes, consisting of the following: 6 pairs tongs, 4 chisels, 6 punches, 4 hammers, 3 sledges, 6 sets swages, 6 sets of fullers, 1 flatter, 1 hardy, and 1 set hammers.

There will be provided and set in the blacksmith shop an approved blacksmith's forge, about 42 inches square by about 25 inches high. Forge to be complete with tuyere, blast gate, pipe connecting as directed, etc., and with a set of tools of assorted sizes consisting of the following: 6 pairs of tongs, 4 chisels, 6 punches, 4 hammers, 3 sledges, 6 sets of swages, 6 sets of fullers, 1 flatter, 1 hardy and 1 set hammers.

One steel blacksmith's anvil of 140 pounds weight, face 14 inches by 4 inches, horn 8½ inches long, with a cutter hole ⅞ inch square.

All tools to be the best design, material, quality and workmanship, to have the latest improvements and all necessary attachments, wrenches and countershafts, and, when required, provision will be made for working them by hand.

The lathes to have a complete set of change gears for cutting U. S. N. standard screw threads, including 11½ threads per inch.

When submitted for approval there must be furnished full descriptions, cuts or drawings showing details of the machine, also the name of the maker, the net weight, the space occupied, and a list of the tools and attachments.

All machines to be furnished complete with a full set of cutting tools and all necessary attachments for miscellaneous work.

The machine tools will be driven by one or more inclosed motors, either directly or through the medium of line and countershafting as approved.

All of the above tools will be subject to the approval of the Bureau of Steam Engineering.

The following are the specifications for the machine tools for the scout cruisers.

MACHINE TOOLS.

There will be provided and installed in the general workshop the following machine tools, viz:

One screw-cutting, back-geared, extension-gap lathe, to swing at least 28 inches over the upper and 48 inches over the lower ways, with not less than 10 feet between the centers, when extended. To have hollow-spindle compound rest, steady and follow rests, power cross feed, taper attachment, countershaft, 2 face plates, 1 large and 1 small chuck, 1 drill chuck and a complete set of at least 16 lathe tools and 7 lathe dogs.

One screw-cutting, back-geared tool-room lathe, to swing at least 14 inches over the ways and not less than 4 feet between the centers. To have quick-change feed gear, power cross feed, hollow-spindle taper attachment, oil pan, 1 large and 1 small face plate, counter shaft, 1 large and 1 small chuck and drill chuck, drawing-in attachment, with complete set of chucks, a set of at least 12 lathe tools and 6 lathe dogs.

One column tool-room shaper of at least 15-inch stroke and not less than 15-inch traverse. To have automatic stop, adjustable table, and graduated swivel vise with jaws for taper work, 24-inch graduated index center and a set of at least 12 shaper tools.

One upright drill, to drill up to 1½ inches, to have at least 14 inches from edge of work, to have not less than 14 inches of traverse of spindle, to be fitted for No. 4 Morse taper, with requisite number of sockets. To have a complete set of twist drills from ¼ to ⅝ inch by 32ds, and from 11/16 to 1½ inches by 16ths. To have automatic and hand-feed, sliding head, counterbalanced spindle, circular table, revolvable by gear and adjustable vertically by gear, automatic stop, tapping attachment, countershaft and a small drill chuck fitted.

One 16-inch sensitive drill, to have a counterbalanced spindle fitted for No. 1 Morse taper, sliding head, table, to have vertical adjustable movement, to drill holes up to ⅝-inch with drill chuck and complete set of drills from 1/16 to ⅝ inch by 32ds.

One universal milling machine with overhanging arm; to have at least 18-inch longitudinal feed of table, 13-inch vertical movement, and 4½-inch traverse. Spindle to be fitted for No. 9 B. & S. taper. To have automatic cross feed, universal chuck, arbor, index, swivel vise, spiral cutting attachment, countershaft, 3 milling cutters, 2 side millers, 6 metal slit saws, 3 angular cutters, 4 end mills and 2 collets.

One combined hand punch and shears with 6-inch shear blades, to cut ¾-inch round iron, shear ⅜-inch steel plate, and punch ⅜-inch holes in ⅜-inch mild-steel plates 4 inches from edge.

One double emery grinder on column, with carborundum wheels, 12-inch diameter by 2-inch face, with countershaft and attachment for surface grinding and with 2 spare wheels.

One, portable cylinder-boring machine with latest attachments for boring cylinders in place. To bore from 6 to 24 inches diameter and to have at least 30 inches travel.

Six machinist's swivel-bottom vises, with jaws at least 5¼ inches wide opening 8 inches, with pipe-vise jaws and copper vise lips.

One approved steel blacksmith's forge, of the portable folding type, with all the necessary tools and fixtures. The blast wheel to be about 12 inches diameter turned by a crank. The

gearing enclosed in a dust-proof case. When set up, the pan to be about 24 inches square. The pan and hood to form a case which, when the forge is dismantled and the parts placed inside, will be a strong, impact, steel box, provided with handles for carrying. This forge will be supplied with a set of tools of assorted sizes, consisting of the following: 6 pairs tongs, 4 chisels, 6 punches, 4 hammers, 3 sledges, 6 sets swages, 6 sets of fullers, 1 flatter, 1 hardy and 1 set hammers.

There will be provided and set in the blacksmith shop an approved blacksmith's forge about 42 inches square by about 25 inches high. Forge to be complete with tuyere, blast gate, pipe connecting as directed, etc. And with a set of tools of assorted sizes consisting of the following: 6 pairs of tongs, 4 chisels, 6 punches, 4 hammers, 3 sledges, 6 sets of swages, 6 sets of fullers, 1 flatter, 1 hardy and 1 set hammers.

In addition to the accessories enumerated above the following are desirable:

For large lathe: Pipe center to fit tail stock, to be used in turning pipe;

Bell center for centering pipes;

Two centering tools, same size as pipe center;

One center grinder to fit tool post head;

Additional lathe tools for special work;

For small late: One steady rest;

One follower set;

Pipe center to fit tail stock;

Bell center, same as pipe center;

Armstrong tool holder;

One center grinder;

Two centering tools, the same size as the pipe centers.

For drill press: One boring bar complete with tapered shank.

Machine tools on board ship are driven by independent electric motors, by a motor driving shafting from which each tool is operated by a belt, and by a small steam engine driving shafting.

The most approved system is considered to be that where each tool is driven by its own motor. This dispenses with the use of shafting and belts, and thus gives greater space and facilities for handling work about the machines.

The following is a list of material for use in work shop:

Castings: Hard cast iron, hollow, of various diameters, for making packing rings, valve-chest liners, etc.

Steel billets: For making crosshead and crank pins of auxiliaries, especially for blowers, ice machines and ash hoists.

Iron bar: Round and flat, of various sizes, for brackets, ties, supports, etc.

Steel bars: For making extension spindles, valve stems and piston rods of auxiliaries.

Tool steel: For making chisels, tools and special wearing parts.

Brass castings, various sizes: For making brass fittings of different kinds.

Brass and Manganese-bronze rods, round and hexagonal; For making brass valve stems, valves, bushings and other fittings.

Sheet brass, assorted sizes: For liners, shims, distance pieces and pipe covering.

Sheet copper: For gaskets and pipe covering.

Spring steel and brass: For springs.

Sheet iron, plain and galvanized: For making covers, tanks, shelves, etc.

Brass, iron and copper piping of various sizes.

Copper tubing: For small steam pipe and gauge connections.

Other accessories to the machine shop are surface plates, usually a small and a large one, plate gages, sets of taps and dies for bolts and nuts, sets of pipe taps and dies, pipe cutters, pipe wrenches, corrugating machine for corrugating copper gaskets, electric and pneumatic drills and hammers, buffing wheel to be used on emery grinder spindle for polishing.

OUTFIT OF TOOLS AND SPARE PARTS.

On commissioning, vessels are furnished by the builders with an outfit of special tools and spare parts. These may be considered as a permanent outfit; additional tools, gear and supplies are furnished from the General Storekeeper of the Navy Yard in accordance with the allowance list of the vessel.

The following are the latest requirements of the machinery specifications for the permanent outfit of tools and spare parts.

The following tools will be provided for the engine and firerooms, viz:

Two ratchets for turning gear.

One wrench for propeller-blade nuts.

One wrench for propeller-shaft nuts.

One set of casehardened wrenches, with racks for each auxiliary engine, to be secured to the bulkhead near its engine.

One set of wrenches complete for each engine and fireroom, fitted for all nuts in the respective compartment, plainly marked as to size and any special use, and fitted in iron racks of approved pattern. The set will include duplicates of all special wrenches found to be necessary in erecting and installing the machinery.

Wrenches for nuts of bolts less than 1 inch in diameter will be finished. For sizes over 2 inches they will be box wrenches, where such can be used. Socket wrenches will be furnished where required. Open-end wrenches will be of forged steel with casehardened jaws; all others will be of forged or cast steel.

A fixed trammel for setting the main valves without removing the covers. The valve stems will be properly marked for this purpose.

Fixed trammels or gauges for aligning crank shafts.

Two tube expanders for evaporator tubes.

One tube expander for feed-heater tubes.

One tube expander for distiller tubes.

A hydraulic jack of approved pattern for withdrawing the bolts of the main-shaft couplings.

One boiler-test pump, with gauge, to test up to 500 pounds;

to be fitted with 10 feet suction and 24 feet pressure hose of sizes to suit pump, with couplings and connections complete.

Steel templets, $\frac{1}{8}$ inch thick, of the bore in propeller hubs.

One hundred condenser-tube plugs, brass, hollow, threaded same as ferrules.

One hundred feed-heater-tube plugs, cast iron, slightly tapered.

Scrapers for brasses, of various sizes and shapes.

Three portable workbenches for firerooms, to be easily taken apart or bolted together. Each to be fitted with Parker's combination vise or equivalent, for holding 3-inch pipe or under; to be 2 feet 6 inches high, 2 feet 6 inches wide, and 4 feet long.

Planks of suitable lengths and dimensions, properly shaped for use as staging about crank pits.

Boxes for cleaning gear, 12 by 6 by 5 inches, to be strongly made of galvanized iron, with vertical partition in middle and with handle in end to draw from shelf; this end stamped "U. S. S.———, Engineer Dept."

Pigeonhole cabinet shelves for stowing the above boxes, to be strongly made of galvanized iron and placed where directed.

Brass lamp trays, about 4 feet by 27 inches, to be about 4 inches deep and provided with perforated division plates in three sections, so as to be easily lifted out. To have draincock and to be secured in shaft alley as directed.

Flanges, for copper steam pipes, of brazing material, as per sketches to be furnished. These flanges to be faced, grooved, and ready for use, except that bolt holes are not to be drilled.

Any special or unusual gear for engine-room telegraphs.

Any special or unusual gear, knuckle joints, bevel gears, etc., for revolution indicator.

Dies for making copper gaskets.

There will be the following tools and stores for the boiler rooms:

Two sets of fire tools for each fireroom. A set of fire tools will comprise two long and two medium-length slice bars, two long and two medium-length fire hoes, one long and one medium-length ash hoe (each hoe with one spare blade), two devil's claws, two pricker bars, two soot hoes, four coal buckets, and two ash buckets.

Racks will be fitted in the firerooms, in convenient places, for holding the fire-tools.

Wrenches for boiler hand-hole nuts, two for each boiler.

Wrenches for superheater hand-hole nuts, one for each boiler.

Wrenches for boiler manhole nuts, one for each boiler.

Suitable wire and bristle brushes for cleaning soot from exterior of tubes.

Tube scrapers with handles, two for each boiler.

Six tube expanders and mandrels, complete, for each size of generating tube.

Four tube expanders and mandrels, complete, for each of the other sizes used in assembling boilers.

Twenty-four plugs for each size of tubes used.

Four plug extractors.

Eight safety-valve gags.

All special tools used in assembling, repair, and cleaning of boilers, as directed.

One turbine tube cleaner for each two boilers, for each size of generating tube, all to be provided with flexible hose.

Four patterns for grate bars.

One pattern for furnace dead plates.

One pattern for fire-door lining.

One pattern for each special form of zinc protector used in boilers.

One set of steel templets, $\frac{1}{16}$ inch thick, for all manhole and hand-hole plate gaskets, each plainly marked to show for which gasket.

Other tools, etc., as may be directed, depending on the type of boiler adopted.

All tools will be conveniently stowed and trammels and gauges will have protecting cases.

SPARE PARTS.

Spare parts will be furnished in accordance with the following tables.

Where machinery not here mentioned or auxiliaries of a different type are installed, spare parts will be furnished in accordance with lists prepared or approved by the Bureau of Steam Engineering.

Spare Parts for Main Engines.—The following spare parts for main engines will be furnished and carried on board :

Part.	Number to be furnished.
Piston rods	1 of each size and pattern.
Valve stems	1 of each size and pattern.
Crosshead slippers	1 of each size and pattern.
Metallic packing, piston rods	All for both engines.
Metallic packing, valve stems	All for both engines.
Eccentric straps, bolts and liners, complete	1 set of each pattern.
Crank-pin brasses, top	1 of each pattern.
Crank-pin brasses, bottom	1 of each pattern.
Liners for crank-pin brasses	2 of each pattern.
Bolts for crank-pin brasses	2 of each pattern.
Crosshead-pin brasses, top	2 of each pattern.
Crosshead-pin brasses, bottom	2 of each pattern.
Liners for crosshead-pin brasses	4 of each pattern.
Bolts for crosshead-pin brasses	4 of each pattern.
Eccentric-rod brasses, top	2 of each pattern.
Eccentric-rod brasses, bottom	2 of each pattern.
Liners for eccentric-rod brasses	4 of each pattern.
Main-bearing brasses, top if fitted	1 of each pattern.
Main-bearing brasses, bottom	1 of each pattern.
Liners for main bearings	2 of each pattern.
Bolts for main bearings	4 of each pattern.
Valve-gear brasses, bushings, etc., not specifically mentioned above	1 of each size and pattern.
Piston rings, complete	1 for each piston.

Part.	Number to be furnished.
Piston-ring springs	All for each piston.
Piston-follower bolts and nuts	25 of each size and pattern.
Cylinder-head bolts	25 of each size and pattern.
Coupling bolts and nuts	3 of each size and pattern.
Split pins, special	2 of each size and pattern.

Boiler Spares.—The following spare parts will be furnished and carried on board:

Part.	Number to be furnished.
Grate bars	Complete for 3 boilers.
Bearers	Complete for 3 boilers.
Dead plates	Complete for 3 boilers.
Furnace door frames, complete	Complete for 3 boilers.
Furnace-door linings	Complete for 3 boilers.
Ash-pit doors	Complete for 2 boilers.
Boiler-manhole plates, with bolts, nuts and dogs	Complete for 2 boilers.
Gauge cocks	Complete for 2 boilers.
Water gauges, complete	Complete for 2 boilers.
Feed check valves	2 to each hand.
Surface blow valves	1 to each hand.
Bottom blow valves	1 to each hand.
Hand-hole plates, with bolts, nuts, and dogs	5 per cent. of all, including superheater.
Special fire brick	10 per cent. of all.
Special fittings	As may be directed.
All tubes of each size and shape, including superheater	8 per cent.
Springs for boiler safety valves	2 of each size.

Miscellaneous Spare Parts.—The following miscellaneous spare parts will be furnished and carried on board:

Part.	Quantity to be furnished.
Condenser tubes of each length	5 per cent.
Feed-water heater tubes	10 per cent.
Distiller tubes	100 per cent.
Evaporator tubes	25 per cent.
Condenser-tube ferrules	10 per cent.
Springs for all safety valves except boiler safety valves	1 of each size, pressure and type.
Springs for all relief valves	1 of each size, pressure and type.

Part.	Quantity to be furnished.
Springs for all reducing valves	1 of each size, pressure and type.
Diaphragms for reducing valves	2 of each size, pressure and type.
Removable seats for all gate valves	2 of each size and type.
Nozzles and flexible metallic tubing of steam tube cleaners	50 per cent.
Zincs not made from standard plates.	100 per cent.
Thrust-bearing horseshoes	All, for either bearing.
Baskets for bilge strainers	100 per cent.
Collar bolts for condenser heads	25.

Spare Parts for Auxiliary Engines.—The following spare parts for auxiliary engines will be furnished and carried on board:

Part.	Circulating-pump engines.	Turning engines.	Workshop engine, if fitted.	Reversing engines.	Forced-draft blower-engines, if fitted.
Connecting rods, of each size and pattern fitted	1	...	...	...	6
Eccentric rods, straps, bolts and nuts, of each size and pattern fitted	1	1	...	...	6
Piston rods and nuts, complete with crossheads and slippers, of each size and type fitted	1	1	...	...	6
Valve stems and nuts, complete, of each size and pattern fitted	1	1	...	...	6
Brasses and bushings, for each one fitted	1	1	1	1	1
Bolts and nuts of crank-pin and crosshead brasses, of each size and pattern fitted, sets	1	1	1	...	6
Metallic packing for all glands, for each size and pattern fitted ... sets	4	1	1	1	12
Cup leathers ... sets	...	...	...	2	...
Packing rings for pistons and piston valves, for each size and pattern fitted	1	...	...	...	6
Springs for pistons and all other parts, for each size and pattern fitted ... sets	1	...	...	..	12

Spare Parts for Pumps.—The following spare parts will be furnished and carried on board:

Part.	Main air pumps.	Auxiliary and dynamo air and circulating pumps.	Feed pumps, main and auxiliary.	Fire and bilge pumps.	Distiller circulating pumps.	Evaporator feed pumps.	Distiller fresh-water pumps.	Oil pumps.	Shaft pumps.	Other pumps.
Steam piston rods and nuts, complete, for each size and pattern fitted...............	1	1	1	1	1	1	1	...	...	1
Pump rods and nuts, complete, for each size and pattern fitted...............................	1	1	1	1	1	1	1	...	...	1
Valve stems and nuts, complete, for each size and pattern fitted.........................	1	1	1	1	1	1	1	...	...	1
Pump valves, with guards, guard bolts, and springs.....	All	All	All	All	All	All	All	All	All	All
Metallic packing, for glands, for each size and pattern fitted, sets.......................	2	2	2	2	2	1	1	1	1	1
Packing rings for steam pistons and piston valves, for each size and pattern fitted..	1	1	1	1	1	1	1	...	...	1
Packing rings for water pistons, for each size and pattern fitted.........................	1	1	1	1	1	1	1	...	...	1
Springs for pistons and all other parts, for each size and pattern fitted, sets........	1	1	1	1	1	1	1	...	...	1
Brasses and bushings for every purpose, for each one or pair, of each size or pattern..	1	1	1	1	1	1	1	1	1	1

Spare Parts for Electric Motors under Steam Engineering.—The following spare parts will be furnished, and carried on board, for each motor:

1 armature, complete with shaft.
1 field coil.
1 set of bearing bushes or linings, complete.
1 set of brush studs, with insulation.
½ set of brush holders.
6 sets of carbon brushes.

1 set of carbon breaks.
2 oil-gauge glasses.
1 set of contacts.
1 set of resistances, completely assembled.
10 inclosed fuses.

Armatures over 120 pounds in weight are to be stowed by supporting the shafts on brackets, secured to the bulkhead, the armature to be incased in a protecting casing of No. 18 U. S. S. G. galvanized steel, in halves, secured by hooks. The journals to be neatly covered with tarred canvas.

Armatures under 120 pounds in weight are to be stowed in plain wooden boxes with hinged covers, hinged hasp, and lock, the armature to be mounted on chocks in such a manner that it can be revolved by hand to permit inspection of all parts. The chocks should be so fitted that the armature can be removed or replaced without damage to it.

Field coils may be stowed in the same box with the armature but in a separate compartment. When the armatures are stowed on brackets, the field coils must be stowed in boxes.

Spare Parts for Air Compressors.—Spare parts for air compressors will be furnished and carried on board. These spare parts will depend upon the type of machine adopted, and a list will be submitted to the Bureau for approval when drawings of air compressor are submitted.

Spare Parts for Ice Machines.—The following spare parts will be furnished, and carried on board, for each ice machine:

Part.	Number to be furnished.
Valve stems for steam valve, compressor distributing valve, compressor cut-off valve, expander distributing valve, expander cut-off valve.	1 of each size and pattern.
Piston rods for steam, compressor, and expander cylinders.	1 of each size and pattern.
Eccentric rods	1 of each size and type.
Crank-pin brasses for steam, compressor and expander cylinders.	1 set for each, of each size and pattern.
Crosshead brasses for steam, compressor and expander cylinders.	1 set for each, of each size and pattern.
Rock-shaft bearing brasses	4 sets.
Valve-rod bushings	10.

Part.	Number to be furnished.
Valves with guards and springs for circulating pump and primer pump.	1 set of each, of each size and pattern.
Springs for steam pistons	$\frac{1}{2}$ set for each.
Cooling coil	1.
Packing for piston rods and valve stems	1 set for each set fitted.
Packing, leather	6 sets for each set fitted.
Thermometers	1 for each fitted.
Wrenches	1 set complete.

Spare Parts not Carried on Board.—The following spare parts, not carried on board and not included in the penalty weight, will be furnished:

Part.	Number to be furnished.
Main propeller shaft	1.
Main-engine crank shaft	1 of each pattern.
Propeller blades	All.
Evaporator steam heads	All.
Boiler tubes, of each size and shape	$\frac{1}{4}$ of total number fitted.
Boiler-tube nipples	$\frac{1}{4}$ of total number fitted.

The packings for compressor and expander cylinders of ice machines will be securely and separately packed in tight boxes with covers and filled with castor oil.

All spare parts will be finished and fitted ready for use, as specified above, and stowed in an approved manner.

Boiler tubes will be securely stowed in racks, or as directed.

Condenser tubes will be packed in boxes. All brass pieces will be stamped with name, and finished pieces, liable to corrode, will be painted with three coats of white lead and oil, and well lashed in tarred canvas, with the name painted on outside.

The spare propeller blades will be of such pattern as may be directed after trial of vessel.

The spare shafts, propeller blades, and such other spares as may be directed, will be packed for storage at a Navy Yard.

The following additional spares may be desirable:

Main Engines.—2 packing rings for each size of balance piston.

1 packing ring and follower for each size of piston valve.

Boilers.—Spare side and cross boxes where water-tube boilers are fitted with these.

Ice Machines.—100 feet galvanized-iron cooling pipe.

The following additional tools may be found desirable:

Set of star wrenches.

Clamp for turning out bottom main-bearing brasses.

TESTS OF MACHINERY AND PIPING.

As a guide to tell what pressures may safely be allowed to be put on various parts of machinery and piping, the following tables giving the latest requirements of the machinery specifications will be valuable. Practically all parts of piping and chambers are fitted with relief valves designed to operate at pressures considerably below that of the test pressure. It is often, however, desirable to test certain parts for strength after being repaired.

Tests of Boilers and Machinery.—The following parts of machinery will be tested *before being placed on board the vessel:*

Part.	Test pressure per square inch above atmosphere. *Pounds.*	Test.
Main boilers	450	By the application of heat to fresh water in the boilers, boilers being quite full. This test optional.
Circulating-pump casings	30	Water pressure.
Condenser shells and water chambers	30	Do.
Cylinders and all parts of auxiliary machinery subject to boiler pressure.	450	Do.
Cylinders, H.P., valve chests and connections.	450	Do.
Cylinders, I.P., valve chests and connections.	200	Do.
Cylinders, L.P., valve chests and connections.	100	Do.
Distiller heads, shells and tubes	50	Do.
Evaporator shells	50	Do.
Evaporator steam heads and tubes	450	Do.
Feed-water heaters when on suction side of pumps.	100	Do.
Feed-water heaters when on discharge side of pumps.	500	Do.

Part.	Test pressure.	Test.
Fittings and connections subject to boiler pressure.	450	Water pressure.
Pumps, feed, water cylinders, valve chests and air vessels.	500	Do.
Pumps, fire and bilge, water cylinders, valve chests and air vessels.	225	Do.
Shaft casings after being fitted on shafts.	30	With boiled linseed oil.
Tanks, air compressor..........................	150	Water pressure.
Tanks, feed, filter, hot well, distiller, reservoir, soda, oil, tallow, waste, store, etc.	10	Do.
Other parts and tests..........................	As directed	As directed.

Parts to be tested will be so placed that all parts are acessible for examination by the inspector during the tests.

Pipes will be tested, by water pressure, before being placed on board, as follows:

Pipes.	Test pressure. *Pounds.*
Air pump, main and auxiliary, suction and discharge..................	50
Bilge discharge..	150
Bilge suction..	100
Bleeder..	450
Blow, boiler..	450
Blow, evaporator..	100
Boiling-out condensers..	450
Circulating pumps, main and auxiliary, suction and discharge......	100
Distiller, circulating..	100
Distiller, fresh water..	100
Drains, crank pit..	50
Drains to traps..	450
Drains, separator..	450
Drainage suction..	100
Escape..	30
Exhaust, auxiliary..	50
Exhaust, branch..	50
Exhaust, dynamo..	50
Exhaust, to atmosphere..	50
Exhaust, to feed heater..	50
Exhaust, main..	50
Feed, discharge..	450
Feed, evaporator..	100
Feed, suction..	100
Fire extinguishing on grates..	100
Fire-main supply connections..	225

Pipes.	Test pressure. *Pounds.*
Fire main to stern tubes	225
Fireroom hydrants	100
Fresh water from ship's side	100
Fresh-water pumping system	100
Indicator	450
Jacket steam	450
Overboard discharge	150
Overflow from feed tanks	30
Pneumatic main	150
Pumping-out boilers	450
Receiver, I.P.	200
Receiver, L.P.	100
Refrigerating plant, air pipes	250
Reserve feed-tank connections	100
Sanitary connections	100
Sea suctions	100
Shaft bilge-pump discharges	150
Shaft bilge-pump suctions	100
Steam, auxiliary	450
Steam, branch	450
Steam, galley	450
Steam, dynamo	450
Steam, main	450
Steam, radiators	150
Steam, from evaporators	100
Steam, to evaporators	450
Steam, to I.P. receiver	450
Steam, to L.P. receiver	450
Steam, to sea valves	450
Vapor	30
Water service	50
Whistle and siren	450
Other pipes	As directed.

Valves, manifolds, and castings for pipes will be tested, by water pressure, before being placed on board, as follows:

Valves, manifolds, and castings.	Test pressure. *Pounds.*
Feed valves and castings	500
Steam valves	450
Exhaust valves	100
Water valves, low-pressure	100
Composition castings for feed	500
Steel castings for steam	450
Composition castings for low-pressure	100

The following parts of machinery will be tested *after being placed on board the vessel:*

	Test pressure per sq. in. above atmos. *Pounds.*	Test.
Main boilers	450	By the application of heat to fresh water in the boilers, the boilers being quite full.
Main boilers and pipe connections	375	By steam, and all leaks to be made tight before parts are clothed and lagged.
Main and auxiliary feed systems as a whole, installed and connected, from and including the pressure parts of the pumps up to the valves on the boilers.	500	Water pressure.
All parts after being secured on board.	Working pressures.	Under working conditions.
Other parts and tests	As requ'd.	As directed.

The circulating pumps will be tested by discharging water under a head of 25 feet, 5 feet of which will be suction.

The suction and discharge pipes, during the test, will be the same size as the nozzles on the pumps.

A four-hour test of evaporating and distilling plant will be made under service conditions, using sea-water feed, and maintaining a density of $\frac{3}{32}$ in the evaporators.

No lagging or covering will be on cylinders, pipes or condensers during the tests.

If india-rubber valves are used, they will be taken at random and must stand a dry-heat test of 270 degrees F. for one hour and a moist-heat test of 400 degrees F. for three hours without injury.

Steam will not be raised in the boilers until after the water test of boilers on board, unless desired for drying or testing joints, for which purpose the pressure will not exceed 10 pounds per square inch.

CHARACTERISTICS OF DIFFERENT METALS.

Different metals vary in their power to resist hammering, bending or any other shock, and a due consideration of these properties will very often avoid a serious breakage.

Cast Iron.—This should never be subjected to heavy blows as it is brittle and will readily crack. It cannot be bent, and if sufficient force is applied it will break. It can stand a crushing stress very well, but no sudden shocks. Cast iron will also crack when subjected to sudden changes in temperature.

Cast Steel.—This will stand sudden blows and hammering and can also be bent to some extent, but is not easily deformed. It has very little flow under stress. It is more brittle and less liable to deformity and bending when subjected to blows than forged steel. Cast steel is very liable to crack when subjected to sudden changes of temperature.

Forged Steel.—This is tough and elastic and will withstand sudden shocks very well, and it can be bent without any great danger of cracking. It will not crack as readily as cast metal when subjected to changes in temperature.

Forged Iron.—This is tough and softer than steel, but not as strong. It can be bent without danger of cracking. Hammering will not cause it to crack, but will easily force it out of shape. Changes in temperature will not cause it to crack readily.

Cast Brass or Composition.—This, like other cast metal, is somewhat brittle. The larger the percentage of copper the less brittle will brass be, and the more will it be able to withstand hammering and bending. It will expand more when the temperature is raised, but will not crack as easily as cast iron when temperature is suddenly changed.

It will be noticed that cast iron should not be struck by blows nor should there be any endeavor to bend it or to bring it into shape by hammering. Cast steel can be bent, but should not be struck, as it may crack; the same may be said of composition.

In striking of metals that are likely to be deformed by the process, use a soft copper maul, and never a hammer of any material harder than that which is struck, since in that case the material struck will be deformed and the hammer retain its shape. When a soft hammer is used it will be knocked out of shape but the part on which it is used will not be deformed.

To prevent edges burring or causing deformity, place pieces of wood against the part struck. The blow will then be distributed over a greater area and the part struck will not be deformed. In handling a delicate piece in a vise, strips of wood or cloth should be used to protect the surface, which might be otherwise dented or grooved by the vise jaws.

WEAR OF METALS ON EACH OTHER.

Whenever metal works on metal nearly all the wear will take place on the softer metal of the two; the softer will thus be worn away and the harder will retain its shape. This fact is often considered in arranging the various parts of machines so that the softer parts can be renewed and replaced. As a rule for wearing surfaces, two different kinds of metal are used. Cast iron will wear well on itself, but cast steel and forged iron will not. Brass will wear well on steel or cast iron.

EXPANSION AND CONTRACTION OF METALS.

All metals in general contract with cold and enlarge when heated, but the relative amount of this extension differs widely for different materials.

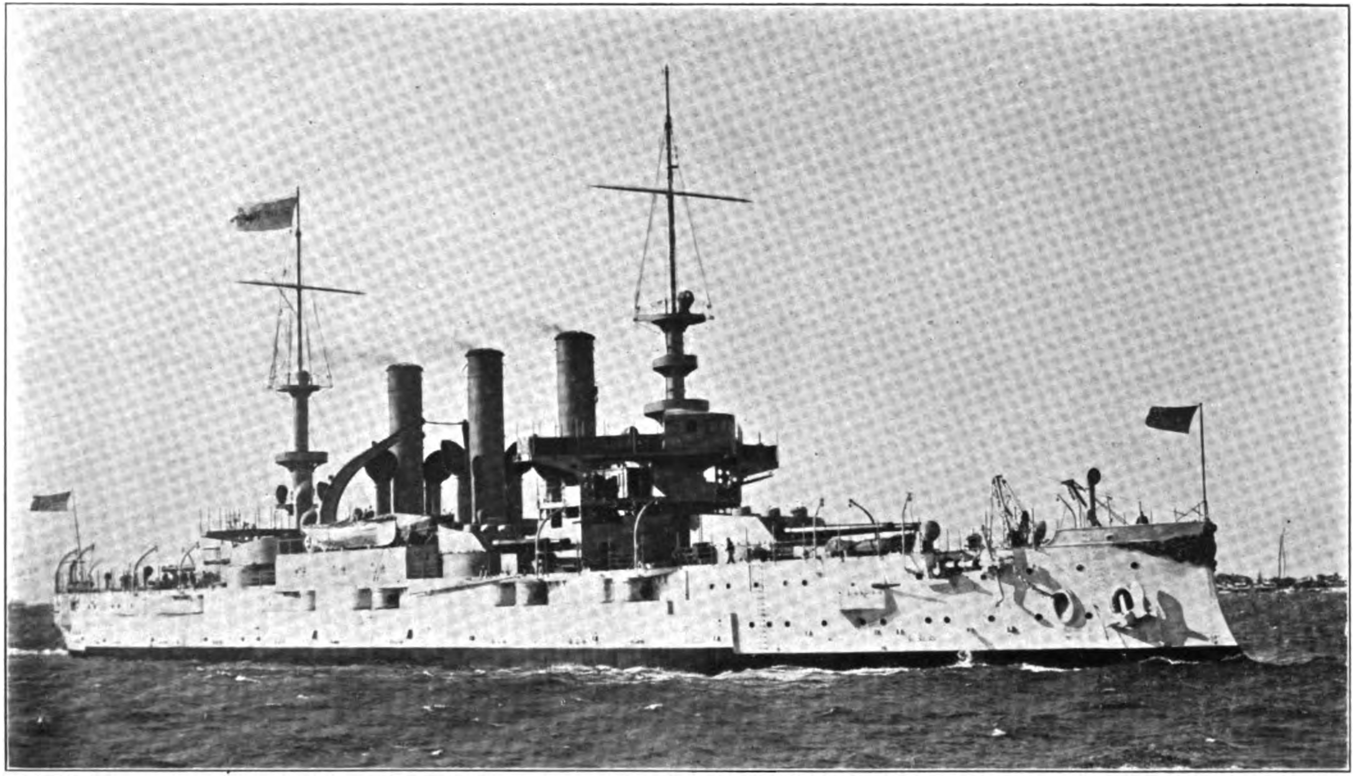
The extension per linear foot in inches for each degree in temperature Fahrenheit of metals ordinarily met with is as follows:

Brass	.0125	Steel, rod	.00763
Cast iron	.0074	Steel, cast	.0072
Copper, from 0° to 212°	.0115	Steel, tempered	.00826
Copper, from 32° to 212°	.00418	Steel, not tempered	.00719
Gun metal	.0127	Zinc, forged	.0207
Iron, forged	.00814	Zinc, sheet	.0196
Iron, forged 0° to 212°	.00788	Zinc, 81 tin	.0179
Iron, forged, 32° to 212°	.00326		

This property can often be made use of to loosen nuts that are frozen, which when heated up expand and can be moved. In connection with this it is well to note the different co-efficients of expansion as shown above.

The property of metals contracting when cooling can be made use of in bringing parts together with great force. If an iron clamp or ring is put on while hot, on cooling it will contract and grip tightly; and a rivet put in hot will draw the plates together. Collars and jackets are often shrunken on by being thus heated and then cooled after being put in place.

Water in freezing expands with great force and will cause the containing vessel to be fractured. Hollow shells have been broken in this way.



U. S. S. "VERMONT."

U. S. S. *VERMONT.*

DESCRIPTION AND OFFICIAL TRIAL.

The *Vermont* (first-class Battleship No. 20) built by the Fore River Shipbuilding Company of Quincy, Mass., is one of three first-class battleships authorized by Congress in an act approved March 3, 1903, and is the second of these three ships to have her trials.

The two sister ships are the *Minnesota,* building at Newport News, Va., and the *Kansas,* building at the New York Shipbuilding Company's Works, Camden, N. J.

The contract for the construction of the *Vermont* was concluded the 20th day of June, 1903, and provided for the delivery of this vessel in forty-two months from that date, that is, on or before December 20th, 1906. The time limit was extended to February 11th, 1907.

The contract price was $4,179,000, of which $1,000,000 was allowed for the building and complete installation of the machinery under cognizance of Bureau of Steam Engineering.

The guaranteed speed of the vessel on trial of four hours' duration was not less than an average of eighteen knots an hour, during which period an air pressure not exceeding one inch of water was allowed in the firerooms, and the vessel weighted to a mean draught of twenty-four feet six inches.

The designed indicated horespower of the propelling engines was to be about 16,500 when making about 120 revolutions a minute with a steam pressure of 250 pounds.

Failure to attain the guaranteed speed of eighteen knots under the stipulated trial and conditions entails penalties at the rate of $50,000 per quarter knot for speed between 18 and 17¾ knots, and $100,000 a quarter knot for speed between 17¾ and 17½ knots, and that if the vessel should fail to exhibit an average speed of at least seventeen and one-half

knots an hour it should be optional with the Secretary of the Navy to reject her or to accept her at a reduced price and on conditions to be agreed upon.

COMPLEMENT.

Admiral,	1	Seaman branch,	401
Chief of staff,	1	Artificer branch,	44
Commanding officer,	1	Engine-room branch,	226
Wardroom officers,	19	Special branch,	17
Junior officers,	20	Messmen branch,	44
Warrant officers,	9	Marines,	72
		Additional for flag,	36
	51		
	840		840
Total,	891		

GENERAL HULL DATA.

Length between perpendiculars, feet and inches	450–00
Projection forward of F. P., feet and inches	6–04
aft of A. P., feet and inches	0–00
Length over all, feet and inches	456–04
of L. W. L., feet and inches	450–00
between perpendiculars, feet and inches	450–00
Breadth, molded, feet and inches	76–05½
extreme, feet and inches	76–10
Depth, molded, main deck side at M. S. (56½), feet and inches	43–01⅜
Draught, trial, feet and inches	24–06
Ratio, length to beam	5.856
Displacement (normal, 24 feet 6 inches draught), trial, tons	16,000
(Estimated 900 tons coal, 50 tons reserve feed water.)	
Area midship section, square feet	1,808
L. W. L. plane, square feet	26,560
wetted surface, square feet	44,500
C. G., L. W. L. plane aft of M. S. (56½), feet	4.4
C. B. above bottom of keel, feet and inches	13–04
aft of M. S. (56½), foot	.765
Transverse metacenter above C. B., feet and inches	18–09½
Longitudinal metacenter above C. B., feet and inches	538–06
Coefficient of fineness, block	.6592
midship section	.9604
L. W. L. plane	.7683
cylindrical	.6873

Shafts incline up and aft at angle of 55 minutes 42 seconds.
Diverge at angle with center line of 25 minutes 41.2 seconds.

Center line at 72: 10 feet 6 inches from center line of ship, 7 feet $\frac{3}{8}$ inch from base line.

Center line 12 inches aft 108: 11 feet 7 inches from center line of ship, 9 feet $4\frac{1}{8}$ inches from base line.

Rake up, .19446 inch per foot; out, .08966 inch per foot.

Screws, diameter, feet and inches	**17–09**
pitch, adjustable 17 feet to 19 feet, feet and inches	**18–00**
area expanded, square feet	**93.0**
projected, square feet	**71.05**
center forward of A. P. (26 inches aft 108), feet and in.	**16–10**
Rudder axis forward of A. P. (3 feet 9 inches aft 108), feet and in.	**15–03**
Strut center line forward of A. P. (20 inches forward 108), feet and inches	**20–08**
Number of frames (spaced 4 feet throughout)	**112**
Watertight compartments	**314**

GENERAL DESCRIPTION.

The hull is made of basic, open-hearth steel, with frames spaced 4 feet apart, except under the engine compartment, where the frames are spaced 2 feet apart.

The inner bottom extends from frame 12 to frame 98. The double-bottom compartments between frames 58½ and 67½ are arranged for use as reserve-feed tanks, and will hold 150 tons of feed water. A cofferdam, 30 inches in width, extends the entire length of the ship about the protective deck, being carried to a height of 3 feet above the protective deck, and is packed with corn-pith cellulose in those parts forward and abaft the midship armor. All unexposed decks throughout the ship are covered with linoleum, except in the magazines and certain storerooms which are fitted with wood shutters, and in rooms, water closets, etc., where tiling is used.

The Superstructure.—The superstructure, built on the main deck, extends from frame 35 to a point 21 inches abaft frame 79, or between the 12-inch turrets; it extends the full width of the ship between the forward and after pairs of 8-inch turrets; it is recessed to make room for these turrets, and is cut away to permit a dead-ahead fire for the forward pair and a dead-astern fire for the after pair of 8-inch turrets. It is also cut away at the corners to give an arc of fire of 135 degrees for the 12-inch guns, forward and aft.

The Bridge Deck.—The bridge deck forms the top of the

superstructure. Over the forward end is built the forward bridge, on which is the chart house, emergency cabin, entrance to conning tower and the torpedo-directing station, which is joined to the forward side of the conning tower. Over the forward bridge is the usual flying bridge, on which are the steering compass, two searchlights and the usual navigating instruments. The standard compass is mounted on top of the chart house.

On the after end of the bridge deck is the after bridge, from which is the entrance to the armored signal tower.

Top Hamper on the Superstructure.—There is one electric boat crane on each side of the ship, of 15-tons capacity, located abreast of the after funnel. The boats, with the exception of two 30-foot whaleboats, 30-foot barge, 30-foot gig whaleboat, and two 30-foot cutters, which are on the main-deck davits, are stowed on the bridge deck. The dinghies are located abaft the after funnel, between the recesses for the after pair of 8-inch turrets, the 16-foot dinghy nested in a 20-foot dinghy on port side near center of ship, and the 14-foot dinghy in the other 20-foot dinghy on starboard side near center line. The other boats are stowed abreast in wake of the middle and after funnels. Beginning outboard, those located under starboard crane are: one 50-foot steam cutter, which is temporarily replaced by a 40-foot steam cutter; one 36-foot steam cutter, and one 30-foot cutter, which is nested in a 33-foot launch. Under the port crane are: one 36-foot steam cutter; a nest of one 30-foot cutter; one 33-foot launch and one 36-foot launch; and a nest of one 30-foot cutter and a 33-foot launch.

Main Deck.—The main deck extends the full length of the vessel, the superstructure partially covering it, as described above. At the bow is a waist plate, 3 feet 6 inches high, extending aft to frame.

Anchors, etc.—There is one anchor crane on each side, for handling Navy-type sheet anchors, bill boards for which are located between frames 9 and 13. There is, however, but one Navy-type anchor provided, which weighs 17,490 pounds, and

which is stowed on the starboard bill board. The tackles for these cranes are operated by electric winches abreast windlass house. There are two hawsepipes on each side, opening through this deck on to the forecastle, from which the chains have a direct lead to the windlass house, which is in front of the forward 12-inch turret and projects above the deck so that the chains enter the front, directly to the wildcat. In the forward hawsepipes are carried two 17,350-pound Baldt stockless anchors. There are also two kedge anchors, weighing 810 and 410 pounds, carried forward, and a 5,655-pound stern anchor and a 4,160-pound stream anchor, carried aft, all stowed against plating of superstructure. Forward, two 20-foot dinghies are stowed for harbor position only and two 30-foot cutters permanently, and aft there are two 30-foot whale boats, a 30-foot barge and a 30-foot gig whale boat.

Within the superstructure, and at its forward end, is the crew's galley and mess issuing room; amidships is the officers' galley, and aft are the general mess pantry, paymaster's and executive offices, battalion lockers and wireless-telegraph room. Drying rooms are located around all funnels.

Gun Deck.—Forward, on this deck, are located the crew's showers and washroom, petty officers' and machinists' water-closets, and the staterooms of the sergeant of marines and master-at-arms. Between the 12-inch barbettes is the 7-inch battery, in which space are located the bakery, printing office, brigs, junior, warrant and wardroom officers' lavatories, baths and shower baths; armory and engineers', navigators' and ordnance officers' offices. Abaft of the after 12-inch barbette are the admiral's, captain's and chief-of-staff's quarters and eight wardroom staterooms.

The admiral's quarters are unusually spacious, and take up nearly the entire after part of this deck.

Due to the installation of 7-inch guns on this deck, the deck height is greater than has been previously customary, being 8 feet 7 inches between decks and 7 feet 9 inches to bottom of beams. This necessitated special arrangements for the hammock hooks, which extend about 7 inches below bottom of beams.

Berth Deck.—On this deck are located, beginning forward, lamp rooms, stowage for oils, paint-mixing rooms, commissary stores and band rooms, the isolation ward, sick bay and its bath, dispensary and operating room. The machinists' and petty officers' quarters and washrooms and the refrigerating plant are abreast the forward 12-inch barbette; adjoining these are the steam laundry, general mess stores and paymaster's issuing room. The firemen's and servants' washrooms are on the center line between uptakes and take up their entire length. Abaft of uptake is the evaporator room. The general workshop is over the engine space, between hatches. Abreast of the engine-room hatches and after barbette are the junior and warrant officers' quarters and mess rooms, on starboard and port sides, respectively. Abaft, are the wardroom officers' quarters; farther aft several showers are provided.

Protective Deck.—This deck extends the full length of the ship and is 3 inches thick on the slopes and 1½ inches on the flat. The cofferdam previously mentioned extends its entire length, and the spaces between the slopes and berth deck are used for stores, except from frame 40½ to 72, where the berth-deck coal bunkers extend to this deck.

Platform Decks and Hold.—There are two platform decks. On these decks are the magazines, storerooms, handling rooms, etc. On the upper platform are the 8-inch ammunition, forward fresh-water tanks, and one dynamo room at each end of the boiler compartments, and the steering gear and engine. On the lower platform are the 7-inch, 3-inch and 3-pdr. magazines and the forward and after torpedo rooms. In the hold are the chain lockers, magazines, 3-inch and 3-pdr., 1-pdr., 30-caliber and saluting powder, for naval-defense mines and torpedo war heads; also the after fresh-water tanks.

ARMOR.

The side belt is a complete water-line belt, 9 feet 3 inches wide amidships, for the length of the lower casemate; forward, this width is reduced to 8 feet and aft to 71 inches; from

frames 24 to 95, or 285 feet in wake of machinery spaces, the thickness is 9 inches, parallel.

Forward and aft of these points the thickness is 7 inches at top, tapering to five inches at bottom, then reduced to five inches constant thickness, then to four inches constant thickness. Top of belt is a straight line 4 feet 3 inches above L. W. L.

Covering the middle length of the ship there is a lower casemate belt of 7-inch armor extending from the top of the heavy side belt to the lower edge of 7-inch gun ports, 2 feet 8 inches above the gun deck. Above this, and extending to the height of the main deck, is a 7-inch belt, which is bent in to form wing plates at the 7-inch gun ports.

Between the gun deck and the protective deck, at frames 24 and 94, there are athwartship casemate bulkheads 6 inches thick; these are outside of the 12-inch barbette. Between the gun and main decks are similar 6-inch bulkheads at frames 28 and 86, connecting to the 12-inch barbettes. These athwartship bulkheads connect the ends of the casemate armor, forming a complete central casemate.

Between the gun deck and main deck, athwartship, splinter bulkheads of nickel-steel are provided for protection of 7-inch battery, each gun being thus in an isolated protected compartment. These are of 60-pound plates and located at frames 36, 50, 57, 64 and 78; those in wake of boiler uptakes not being continuous, those at frames 36 and 78 being continuous. A continuous fore-and-aft splinter bulkhead of 80-pound nickel-steel plate is fitted between 12-inch barbettes. Armored hinged doors are fitted in splinter bulkheads where necessary to provide access.

The 12-inch barbettes are 10 inches in thickness at front and 7½ inches thickness at back, from the top to the gun deck, and 6 inches between gun deck and protective deck. The 12-inch turrets are 12 inches thick on slope, remainder being 8-inch, top plate 2½ inches.

The 8-inch barbettes are 6 inches thick at front and 4 inches thick at back, with tubes from main to gun decks of

3¾ inches and from gun to protective deck of 3 inches thick.

The 8-inch turrets are 6½ inches thick on slope, remainder 6 inches; top plates 2 inches thick.

The conning tower is 9 inches thick and fitted with a swing door. The armored tube for communication is 6 inches thick. The signal tower aft is 6 inches thick.

BATTERY.

The 12-inch guns are mounted in pairs in two elliptical, electrically-controlled balanced turrets, one forward and one aft on center line, each having an arc of fire of 270 degrees.

The 8-inch guns are mounted in pairs on four elliptical turrets, located two on each side and about equally spaced between the two 12-inch turrets. Each pair have an arc of fire of 145 degrees, the forward pairs from 90 degrees forward to 55 degrees abaft beam; after pairs, vice versa.

The 7-inch guns, twelve in number, are mounted singly on pedestal mounts with shields, in the central armored casemates on gun deck, each having an arc of fire of 110 degrees.

The secondary battery consists of twenty 3-inch R.F., twelve 3-pdr. semi-automatic Hotchkiss, two 1-pdr. H.R.F. automatic, two 30-caliber Gatling machine guns. Four 21-inch submerged torpedo tubes are provided for, two forward and two aft. There are two Barr and Stroud range finders, one in each upper military top.

The following table gives the location and train of all guns, viz:

Four 12-inch B.L.R., 135 degrees port and starboard of center line of ship.

No. 1, forward turret, right hand.
2, forward turret, left hand.
3, after turret, right hand.
4, after turret, left hand.

Eight 8-inch B.L.R.

No. 1, forward starboard turret, right hand, 90 degrees forward, 35 degrees abaft beam.
2, forward starboard turret, left hand, 90 degrees forward, 55 degrees abaft beam.

3, forward port turret, right hand, 90 degrees forward, 55 degrees abaft beam.
4, forward port turret, left hand, 90 degrees forward, 55 degrees abaft beam.
5, after starboard turret, right hand, 90 degrees aft, 55 degrees forward beam.
6, after starboard turret, left hand, 90 degrees aft, 55 degrees forward beam.
7, after port turret, right hand, 90 degrees aft, 55 degrees forward beam.
8, after port turret, left hand, 90 degrees aft, 55 degrees forward beam.

Twelve 7-inch R.F., on gun deck.

Nos. 1 and 2, frame 31, 90 degrees forward, 26 degrees abaft beam.
3 and 4, frame 39, 73½ degrees forward, 42½ degrees abaft beam.
5 and 6, frame 53, 70 degrees forward, 46 degrees abaft beam.
7 and 8, frame 61, 46 degrees forward, 70 degrees abaft beam.
9 and 10, frame 75, 42½ degrees forward, 73½ degrees abaft beam.
11 and 12, frame 83, 26 degrees forward, 90 degrees abaft beam.

Twenty 3-inch 50-caliber R.F.

Nos. 1 and 2, gun deck, frame 16½, 90 degrees forward, 30 degrees abaft beam.
3 and 4, gun deck, frame 101, 30 degrees forward, 90 degrees abaft beam.
5 and 6, gun deck, frame 109, 30 degrees forward, 90 degrees abaft beam.
7 and 8, main deck, frame 52½, 60 degrees forward, 60 degrees abaft beam.
9 and 10, main deck, frame 57, 60 degrees forward, 60 degrees abaft beam.
11 and 12, main deck, frame 61, 60 degrees forward, 60 degrees abaft beam.
13 and 14, upper deck, frame 36½, 90 degrees forward, 10 degrees abaft beam.
15 and 16, upper deck, frame 50, 90 degrees forward, 10 degrees abaft beam.
17 and 18, upper deck, frame 77½, 10 degrees forward, 90 degrees abaft beam.
19 and 20, forward bridge, frame 37, 90 degrees forward, 25 degrees abaft beam.

Ten 3-pounder semi-automatic Hotchkiss.

Nos. 1 and 2, upper deck, frame 39½, 45 degrees forward, 45 degrees abaft beam.
3 and 4, upper deck, frame 75, 45 degrees forward, 45 degrees abaft beam.

5 and 6, forward bridge, frame 40½.
7 and 8, after bridge, frame 74½.
9 and 10, after bridge, frame 77½.
11 and 12, after bridge, frame 79.

Two 1-pounder H.R.F., automatic.

Nos. 1 and 2, upper deck, frame 64½.

Two machine guns, 30-caliber Gatling.

Nos. 1 and 2, forward bridge, frame 35.

Four torpedo tubes, 21 inches, 2 forward, 2 aft.

AMMUNITION SUPPLY.

Ammunition hoists are provided in number and speed designed to obtain as far as practicable a full supply of ammunition necessary for all guns firing at maximum speeds. In addition to the usual hoists in 8-inch and 12-inch turrets, there are twelve 7-inch hoists capable of handling either shell or charges, and fourteeen combined 3-inch, 3-pdr. and 1-pdr. hoists. All hoists not within the armored citadel are protected with 80-pound nickel-steel plates. In order to supply ammunition to the hoists located along the ammunition passages on each side of lower platform deck, four continuous-motion ammunition-belt conveyors, each about 80 feet in length, are provided. These convey the shell and charges from the magazines located forward and abaft the machinery spaces on that deck.

These conveyors are so designed as to deliver ammunition to any of the hoists, the two forward ones supplying forward hoists, and two after ones the after hoists. They can also be operated in reverse direction to take ammunition into magazines. The shells are carried to the conveyor loading table from the shell room by means of trolleys on overhead rails. Charges and fixed ammunition are carried to conveyor by hand. The magazines for 8-inch and 12-inch ammunition are located on upper platform deck forward and abaft the machinery spaces. The 8-inch ammunition is carried from the magazines to base of turrets through the passages on differential blocks running on overhead trolleys. The 12-inch

magazines are located adjacent to the handling rooms for those turrets.

All the 7-inch hoists combined can hoist all the 7-inch ammunition in about forty-one and a-half minutes.

Hoists 5, 6, 19 and 20, assisted by 23 and 24, can hoist all 3-pdr. and 1-pdr. ammunition in about eighteen minutes.

Hoists 11, 12, 15 and 16, and 19 and 20, assisted by 23 and 24, can hoist all 3-inch, 3-pdr. and 1-pdr. in about twenty-three and one-half minutes.

About 10 per cent. of 7-inch shells will be stowed near the guns.

Hoists 1 and 2 for 3-inch only. Hoists 5, 6, 19, 20, 23 and 24 will be assisted by hoists 11, 12, 15 and 16, the ammunition being whipped or passed from main to upper deck.

Hoists 25 and 26 to hoist 3-inch ammunition only.

WATERTIGHT COMPARTMENTS.

CAPACITY OF DOUBLE-BOTTOM COMPARTMENTS BELOW WATERTIGHT LONGITUDINALS.

Compt.	Gals.	Tons F.W.	Tons S.W.	Compt.	Gals.	Tons F.W.	Tons S.W.
A-94	12,970	48.2	49.54	B-89	6,248	23.2	23.88
A-95	16,912	62.8	64.60	B-90	5,359	19.9	20.45
A-96	18,139	67.4	69.28	B-91	5,359	19.9	20.45
A-97	14,677	54.5	56.05	B-92	4,471	16.6	17.05
A-98	18,041	67.0	68.88	B-93	4,471	16.6	17.05
A-99	24,234	90.0	92.54	C-95	23,991	89.1	91.68
B-80	19,441	72.2	74.28	C-96	19,926	74.0	76.08
B-81	16,614	61.7	63.48	C-97	19,899	73.9	75.97
B-82	23,588	87.6	90.08	C-98	19,360	71.9	73.97
B-83	20,384	75.7	77.85	C-99	18,714	69.5	71.51
B-84	17,071	63.4	65.20	D-96	18,256	67.8	69.74
B-85	24,018	89.2	91.74	D-97	17,017	63.2	65.00
B-86	20,681	76.8	78.97	D-98	17,502	65.0	66.82
B-87	17,234	64.0	65.82	D-99	17,754	66.3	68.27
B-88	6,248	23.2	23.88	Total.	468,579	1,740.6	1,790.11

RESERVE-FEED TANKS.

B-94	5,804	21.55	22.20	B-97	4,982	18.50	19.06
B-95	5,804	21.55	22.20	B-98	4,151	15.42	15.88
B-96	4,982	18.50	19.06	B-99	4,151	15.42	15.88
				Total.	29,874	110.94	114.28

CAPACITIES OF TRIMMING TANKS.

A-1	4,977	18.22	18.74	D-10	14,848	55.13	56.71
A-2	7,764	28.83	29.65	D-11	10,352	38.44	39.54

CAPACITIES OF FRESH-WATER TANKS.

Tank.	Gals.	Tons.	Tank	Gals.	Tons.
22-23 inboard, 1 P. 1 S.	1,517	5.66	98-99 C. L. of ship	1,868	6.97
21-22 middle, 1 P. 1 S.	937	3.50	37-38 gravity	800	2.98
21-22 outboard, 1 P. 1 S.	844	3.15	39-40 gravity	800	2.98
22-23 outboard, 1 P. 1 S.	1,754	6.54			
98-99 1 P. 1 S.	2,040	7.61	Total	10,560	39.39

COFFERDAMS.

	Cubic feet.		Cubic feet.		Cubic feet.
A-100	329	A-127	352	D-118	404
A-112	188	B-102	318	D-108	874
A-113	188	B-103	318		
A-114	289	C-104	342	IN BUNKERS.	
A-115	289	C-105	342		
A-116	390	C-106	288		
A-117	390	C-107	288	B-104	316
A-118	389	D-109	296	B-105	316
A-119	389	D-110	296	B-106	341
A-120	383	D-111	375	B-107	341
A-121	383	D-112	375	B-108	370
A-122	370	D-113	394	B-109	370
A-123	370	D-114	394	B-110	370
A-124	302	D-115	635	B-111	370
A-125	302	D-116	635	B-112	398
A-126	352	D-117	404	B-113	398

There are 314 watertight compartments, of which 217 are below the protective deck, and 97 above the protective deck. They are distributed as follows, viz:

On berth deck, 28 compartments; on protective decks, 69 compartments; on upper platform deck, 50 compartments; on lower platform deck, 31 compartments; in hold, 67 compartments; in inner bottom, 69 compartments.

Compartments A92 to A99, B66 to B99, C89 to C99, D90 to D99, all inclusive, are double-bottom compartments, of which the following are wing compartments under coal bunkers and outboard sides of boiler rooms, viz: A86 to A93, B66 to B79, B88 to B93, C89 to C94, D90 to D95, all inclusive; they form the upper double bottom, between the armor and the sixth longitudinal, which is watertight. The remaining double-bottom compartments are central compartments, extending from the sixth (watertight) longitudinal on one side to the sixth longitudinal on the other side of the keel and non-watertight through the central line.

Double-bottom compartments B94 to B99, inclusive, are fitted for reserve-feed tanks, and compartments B88 to B93, inclusive, A1, A2, D10 and D11 are trimming tanks. C95

and C96 are under the engine rooms. The tables on preceding page give the capacities and the uses of various compartments fitted for special purposes.

GENERAL NOTES.

Coaling Arrangements.—The coal capacity is 2,415.1 tons, of which 718 tons are carried above the protective deck and 364 tons in the transverse bunkers at the ends of the boiler space. This is handled by means of tripod coaling booms; eight of these coaling booms are located on the main deck, four forward and four aft, and are arranged to lower coal bags through combination skylight and coaling hatches and companion and coaling hatches. Four coaling booms are located on upper deck, and are arranged to lower through combination skylight and coaling hatches and trunks. The coal is handled on the gun deck, where it is distributed to 24-inch scuttles and trunks leading to the various bunkers. The whips are operated by electric winches, two forward and two aft on main deck, and two on upper deck.

CAPACITIES OF COAL BUNKERS.

LOWER BUNKERS.

	Cu ft.	Tons.		Cu. ft.	Tons.
B-7	4,967	115.5	B-20	3,448	80.2
B-8	5,619	130.7	B-21	3,317	77.2
B-9	2,335	54.3	B-22	3,317	77.2
B-10	2,335	54.3	C-5	2,152	50.2
B-11	2,290	53.3	C-6	2,935	68.2
B-12	2,290	53.3	C-7	2,244	52.2
B-13	3,102	72.2	C-8	2,244	52.2
B-14	3,102	72.2	C-9	2,529	58.8
B-15	3,437	79.9	C-10	2,529	58.8
B-16	3,437	79.9	C-11	2,300	53.5
B-17	3,455	80.4	C-12	2,175	50.6
B-18	3,455	80.4			
B-19	3,448	80.2	Below protective deck,		1,685.7

ABOVE PROTECTIVE DECK.

B-118	2,692	62.6	B-126	3,288	76.5
B-119	2,692	62.6	B-127	3,288	76.5
B-120	3,024	70.3	B-128	3,356	78.0
B-121	3,024	70.3	B-129	3,356	78.0
B-122	3,324	77.3			
B-123	3,324	77.3			729.4

Total, all bunkers, 2,415.1 tons.

VENTILATION.

The ventilation of the ship is on the plenum, or supply system, except in the special cases of sick bay, waterclosets, dynamo rooms and steering-engine rooms. The ventilation blowers are electrically driven and the general arrangement is in accordance with recent Navy practice. To avoid as far as possible the piercing of watertight bulkheads, the ventilating system is subdivided into a number of small systems. Forced ventilation is provided for all quarters, living spaces, passages, storerooms and magazines below the gun deck, also for all spaces over boilers and around magazines, and for all wash-rooms, closets and similar enclosures above the gun deck.

FIRE PLUGS.

There are sixty-one fire plugs in the ship, distributed as follows, viz:

Hold	5	Main deck	12
Lower platform	4	Upper deck	6
Upper platform	4	Bridge, forward	1
Berth deck	12	Bridge, aft	1
Gun deck	16		61

MASTS AND SPARS.

There are two military masts. The fore mast is at frame 36, the main mast at frame 76. Each mast has an upper and a lower military top and a signal yard. The main mast has a lower searchlight platform, below the lower top. On the fore mast, just above the signal yard, is a lookout platform, fitted with guard rails and in communication with the bridge. The main mast is fitted with a small monkey gaff, and the wireless telegraph connections will be fitted on this mast.

Heights:
Above designed L. W. L., 24 feet 6 inches forward and aft.
24 feet 6 inches W. L. is 24 feet 4$\frac{5}{8}$ inches above base line.
Masts, fore and main:

Truck light, top of, feet and inches	**131-09$\frac{1}{4}$**
Ranging platform floor, top of (forward only), feet and inches	**114-02$\frac{11}{16}$**
Searchlight platform, feet and inches	**75-07$\frac{5}{8}$**
Lower searchlight platform (aft only), feet and inches	**42-07$\frac{5}{8}$**

Masts, fore and main:

Upper top, floor (forward only), feet and inches	64–07⅝
Lower tops, floor, feet and inches	54–07⅝
Bridge at 38 center line (forward), feet and inches	34–09⅛
Flying bridge at 38 center line, feet and inches	44–07⅛
side, feet and inches	43–09⅝
After bridge at 74 center line, feet and inches	34–06⅜
side, feet and inches	33–08⅞
Top of waist plate at stem, feet and inches	24–00
beading at stern, feet and inches	20–09⅛
main deck planking at stem, feet and inches	20–10
stern, feet and inches	20–04⅛
flat armor deck, feet and inches	3–00

POWER WATERTIGHT DOORS.

There are forty-two doors and five armored hatches, which are operated by electrical power. They are of the long-arm system, similar to those installed on previous ships. They are so designed as to be operated on the spot by power or hand gear from either side of the bulkhead or deck. They can also be operated simultaneously by power from the emergency-control station located on forward bridge. Electrically-operated solenoid whistles are provided as a warning when doors are to be so closed. During any period of emergency operation any individual door or hatch can be operated by power or hand, and after such operation during the period of emergency operation the closing repeats itself automatically.

DRAINAGE AND FLOODING SYSTEMS.

The main drain, 15½ inches inside diameter, is fitted on starboard side between the inner bottom and boiler and engine-room floors and has a suction in each engine and boiler room above inner bottom; the valve for each suction is operated from compartment which it drains and also in deck plates on berth deck. The main drain is connected directly with the main circulating pumps, and has an auxiliary connection to the construction and repair drainage manifold near each fire and bilge and auxiliary feed pump. These drainage manifolds are connected to steam engineering pump manifolds.

The secondary drain, 5½ inches diameter, is run on port

side throughout length of machinery spaces and is connected to each construction and repair drainage manifold above mentioned. Branches are led from the secondary drain to bilge wells located in after ends of boiler and engine compartments, those on port side leading directly into drain, and those on starboard side leading through the manifold suctions to bilges forward and abaft machinery spaces, to shaft alleys and double-bottom compartments, are led to the drainage manifolds. All valves on secondary drain are operated 30 inches above floor plates.

Double-bottom compartments under machinery spaces, except those used for reserve-feed water, are provided with means both for drainage and flooding; outside these spaces with mains for drainage only, except for those outboard of sixth longitudinal, where piping is to be fitted for either drainage or flooding. In general, compartments not provided with other means of drainage will be drained by means of hose led through doors or hatches.

The fire mains are of 6-inch copper pipe, sabined and run continuously through machinery spaces on each side of ship, directly under protective-deck beams. The port and starboard systems are cross connected at ends of machinery spaces, forming a circuit to which all fire and bilge pumps are connected direct. Risers from main are fitted to supply necessary plugs in each compartment above. Forward and aft of machinery spaces, throughout magazine spaces, a single main is led from the cross connection to supply risers and plugs in those localities. The fire main is also connected to the magazine-flooding system on upper platform. It has also by-pass connections to the flushing system, the full diameter of the latter.

The flushing system is 4-inch copper pipe, sabined and installed under gun-deck beams. This supplies salt water to all spaces throughout ship where required. On upper platform, forward, two separate motor-driven centrifugal pumps are provided for flushing and providing salt water for showers, washrooms, heads, etc.

The fresh-water service throughout the ship is provided by means of a gravity tank located on upper deck and of 1,600 gallons capacity. Two electrically-operated, reciprocating fresh-water pumps are provided, one aft and one forward, for general service of all tanks.

BOATS.

The following table gives the number and sizes of the boats, with location of each and weight carried by each.

	Men.		Water and provisions.		Weight of each boat, stores and men.	Location on bridge deck.	Frames.
	Each.	Total.	Each.	Total.	*Lbs.*		
1 50-ft. steam cutter (40-ft. steam cutter temporarily fitted).	...	60	...	600	19,680	Starboard aft	52–63
2 36-ft. steam cutters	53	106	530	1,060	16,961 each	P. and S. aft	53–62
1 36-ft. launch	...	78	...	780	9,953½	Port aft	53–62
3 33-ft. launches	64	192	640	1,920	8,470¼ each	1 starb. aft 2 port aft	54–63 53–63
5 30-ft. cutters*	45	225	450	2,250	5,624½ each	1 starb. aft 2 port aft	54–62 54–62
1 30-ft. barge (davits, main deck)	...	37	...	370	5,200	Starboard aft	96–103
2 30-ft. whale boats (davits, main deck)	29	56	280	560	4,802½ each	1 port aft 1 starb. aft	76–84
1 30-ft. whale boat (gig, main deck)	...	20	...	200	4,471½	Port aft	96–103
2 20-ft. dinghies	12	24	120	240	2,079½ each	1 starb. aft 1 port aft	65–71
1 16-ft. dinghy	...	5	...	50	1,324¾	Port aft	65–71
1 14-ft. dinghy	...	4	...	40	1,029¾	Starboard aft	67–71

* The sea position for two of these cutters is on davits, main deck, between frames 30 and 38.

ANCHOR WINDLASS.

There is one American Ship Windlass Co.'s steam windlass with four wildcats, capable of hoisting two 17,600-pound anchors at the same time. It is of the worm-gear type, operated by a vertical shaft, which is connected to the engine on gun deck by bevel gears. The engine is designed to work at 150 pounds steam pressure.

ELECTRIC PLANT.

There are eight 100-kilowatt generating sets of 125 volts pressure at the terminals. The generators were supplied by the General Electric Co., and are of the direct-current compound-wound, multipolar type. The wiring is on the two-

wire system. These eight machines are placed in two dynamo rooms, four in each room. The dynamo rooms are situated under the protective deck, one just forward of the boiler compartments, the other just abaft the boiler compartments, on the upper platform. Steam is supplied to the generators directly from the auxiliary steam pipe; two dynamo condensers, one in a pocket just under each dynamo room, are provided for taking the exhaust steam of only the dynamos; these condensers are so connected that the dynamo exhaust, forward or aft, may be led into either dynamo condenser or into either auxiliary condenser in the engine room.

Dynamo Engines.—The engines are of the vertical, cross-compound, General Electric Co.'s type; all the working parts are enclosed and lubricated under pressure; the introduction of oil into the cylinders is guarded against by fitting a top plate on the enclosing covering, with an oil wiper around valve stems and stuffing box around piston rod, in which are soft-packed stuffing boxes for the rods and valve stems to work through.

Dynamo engines, 100-kilowatt, number of	8
Diameter of cylinders, H.P., inches	10
L.P., inches	18
Stroke, inches	10
Diameter of piston rods, H.P., inches	2.25
L.P., inches	2.25
Revolutions per minute, full load	350

Motors.—All the auxiliary machinery outside the engine and firerooms, except the ice machine, the anchor engine, the forced-draft blowers and the steering engine and evaporator pumps, is driven by electric motors. The following is a list of the motors installed, viz:

No.	*H.P.*	*Winding.*	*Use.*
42	1	Comp.	For operating power watertight doors.
5	1	Comp.	For operating power watertight hatches.
2	50	Series	Hoisting motors for boat cranes.
2	30	Series	Rotating motors for boat cranes.
6	30	Comp.	For deck winches.
26	3	Shunt	For endless-chain ammunition hoists.
4	3.5	Shunt	For ammunition whips.

No.	*H.P.*	*Winding.*	*Use.*
4	4	Shunt	For endless-chain ammunition conveyors.
4	25	Shunt	For turning 12-inch turrets.
8	15	Shunt	For turning 8-inch turrets.
32	Var.	....	For ventilating fans.
1	6	Shunt	For operating steam-laundry machinery.
2	2	Shunt	For operating fresh-water pumps.
2	5	Shunt	For operating flushing pumps for head.
2	1.5	Shunt	For drainage pump for firemen's washroom.

In addition to the above there are the following:

8 portable ventilating sets of 1/4 H.P. each.

58 desk-and-bracket fans of 1/12 H.P. each.

8 bracket fans of 1/6 H.P. each.

Motor Generators.—Turret-Control System.—The turrets are controlled on the Ward-Leonard system. The current for driving the turret-training motors and the gun-elevating motors, both 12-inch and 8-inch, is furnished by motor generators in each turret, which in turn are driven by current from the main generators. With this arrangement one main generator may be used to furnish current for both turret power and lighting at the same time, without having the flickering of lights due to variation in the load when working the turrets.

For each 12-inch turret there is provided a 25-kilowatt motor generator for training horizontal train turning, and for each 12-inch gun, an 8-kilowatt motor generator for elevating. For each 8-inch turret there is provided a 15-kilowatt motor generator for training horizontal train turning, and for each 8-inch gun a 3½-kilowatt for elevating.

Speed regulation is obtained by varying the field of the generator and motor generator up to a certain limit, and beyond that by varying the field of the training or elevating motor.

Lighting Output.—The output for lighting supplies 1,100 16-candlepower incandescent lamps; an additional 100 lamps will probably be provided; also ten 3-ampere arc lights; two 60-inch searchlights (of about 42,000 candlepower each) and four 30-inch searchlights (of about 20,000 candlepower each); two night-signaling sets, two truck lights; two diving lanterns, with eight 150-candlepower lamps.

System of Distribution.—The wiring is on the two-wire feeder system. In each dynamo room there is a generator switchboard to which the four machines may be connected. On the opposite side of the dynamo room, in a separate watertight compartment, there is a distribution board. The forward distribution board feeds all circuits leading to outlets forward of the amidship section of the ship; the after distribution board feeds all circuits leading to outlets abaft the amidship section.

Either distribution board may be connected with either generator switchboard; thus, any generator may be connected with any circuit, lighting, power or turret.

The following circuits are provided with feeders from each distribution board and with double-pole throw-over switches, allowing the use of either feeder; this duplicates the feeders to these circuits, viz:

All lighting circuits in firerooms.

All lighting circuits in engine rooms.

Each circuit feeding motor generator.

Each circuit feeding turret-ammunition hoist and training motors and miscellaneous turret power.

Turret-Hoist Indicators.—A 16-candlepower incandescent-light signal is provided for signaling from the handling room when the ammunition-lift car is loaded and ready for hoisting.

A 16-candlepower incandescent-light signal, automatic in operation, is provided for indicating when the car of the turret-ammunition lift arrives within 42 inches of the extreme lower position. These lamps are, one green and one red.

Interior Communication.—A complete central-station system of telephones is installed. There are thirty-five telephones altogether, including two in central station. There are 139 leads of voice tube.

The telephones are a special type of loud-speaking instrument, made by the Cory Company. With these instruments three wires are required for each instrument instead of six wires, as required by the Bell system. Those instruments

located in stations where they will be subject to exposure, moisture or rough usage, or where there is a great amount of noise, are of watertight type. There are two ear pieces which encircle the ear and exclude external sounds. The instruments are said to work very satisfactorily, even in exposed places in a high wind.

Warning Signals for Watertight Doors.—A system of whistles, operated by solenoids, is installed to give warning of the closing of the power watertight doors. The system for closing watertight doors and hatches is known as the "Long-arm Door System," the doors being operated by electric motors.

Additional Interior Communication.—The usual installation of call bells, fire alarm system, general alarm gongs, boat hour gongs, shaft indicators, steering telegraphs and indicators, engine-revolution telegraphs and indicators, helm indicators, electric gong, battery-control and torpedo-firing apparatus.

PROPELLING MACHINERY.

The propelling engines are right and left hand, placed in watertight compartments abreast each other, separated by middle-line bulkhead. There are two 4-cylinder triple-expansion engines, vertical, inverted cylinders, direct action, each with high-pressure cylinder 32½ inches, intermediate-pressure cylinder 53 inches and two low-pressure cylinders 61 inches, the stroke of all being 48 inches. The engines are of Bureau of Steam Engineering design. They are designed to develop 16,500 horsepower when making about 120 revolutions for a speed of 18 knots per hour, the maximum piston speed in no case to exceed 1,000 feet per minute, and no live steam to be used in the receiver in any full-power trial. They are arranged for *outboard-turning* propellers when going ahead.

The order of the cylinders, beginning forward, is: forward low pressure, high pressure, intermediate pressure and after low pressure. The forward low and high-pressure cranks are opposite, and also the intermediate and after low-pressure

cranks, the second pair being at right angles with the first, the sequence of cranks thus being: high pressure, intermediate pressure, forward low pressure and after low pressure. The cylinder castings are of cast-iron with cast-iron liners in the barrels. The cylinder liners make steam-tight joints with the castings both at the top and the bottom, and the space between the liners and casing forms the steam jacket for the cylinder barrel. All cylinder covers except the H.P.'s are also jacketed.

The engine framings are of forged-steel cylindrical columns, twelve in all, supporting the cylinder crosshead guides, etc. They are flanged top and bottom and secured to the cylinders and bed plates, and have enlarged facings for the athwartship and longitudinal tie-rod braces. The columns are tied fore and aft horizontally by heavy cast-steel eye braces of forged steel and by diagonal rods, and are also strengthened athwartship by forged steel X-frames from top to bottom of columns. The bed plates are of cast steel supported on the keelson plates worked in the hull. The forward L.P. and H.P. engines are secured together by flanges on the sides, as are also the after L.P. and M.P. cylinders. The forward and after pair of cylinders have removable chocks fitted to lugs on H.P. cylinder casing to prevent side motion while allowing expansion in a vertical and fore-and-aft direction. The pairs of cylinders of one engine are secured across to those of the other engines by tie rods through the middle-line bulkhead and also by tie rods through the forward and after engine-room bulkheads.

The engine bed plates are all cast steel and are made in three sections for each engine, bolted together at flanges. Each section carries two main bearings. The bed plates are secured to the keelson plates worked in the hull. The H.P. cylinder has one piston valve. Each of the other cylinders has two, all equal in diameter. The L.P. and M.P. valves differ only in length. The main valves are all of single-ported piston type, built up with top and bottom cast-steel heads and a steel-tube distance piece between the heads. Each

head is fitted with a cast-iron packing piece and cast-steel follower. A balance piston fitted at the top of each steam valve and working in a small cylinder in top of valve-chest cover is provided to relieve a part of weight of valve gear. The valve gear is of the Stephenson type with double-bar links, fitted with adjustable cut-off blocks by which cut-off may be adjusted between the limits of .5 and .76 stroke. The valve gears of each engine are handled by a vertical steam-reversing engine with oil-controlling cylinder secured to the H.P. cylinder case. A reversing shaft is carried in bearing brackets secured to the front of the engine columns, and provided with an arm for each link, fitted with adjustable cut-off block and two arms for reversing-engine connecting rods. Reversing engine is controlled by a lever at the working platform. An 11-inch balanced throttle valve is provided for each engine and operated by a hand wheel at the working platform.

DATA FOR CYLINDERS.

1 H.P., diameter, inches	$32\frac{1}{4}$
1 I.P., diameter, inches	53
2 L.P., diameter, inches	61
Stroke of all pistons, inches	48
Volume swept by H.P. piston per stroke, cubic feet	22,470
I.P. piston per stroke, cubic feet	60,709
L.P. piston per stroke, cubic feet	80,604
Ratio of area I.P. to H.P. piston	2.659 to 1
H.P. to L.P. piston	7.046 to 1
Cylinder walls, thickness, H.P., I.P. and L.P., inches	$1\frac{1}{8}$, $1\frac{7}{16}$, $\frac{5}{16}$
liners, thickness, H.P., I.P. and L.P., inches	$1\frac{1}{2}$, $1\frac{5}{16}$, $1\frac{1}{4}$
Valve-chest liners, thickness, H.P., I.P. and L.P., inches	$1\frac{1}{4}$, $1\frac{1}{4}$, $1\frac{1}{4}$
Jacket space, inch	$\frac{3}{4}$

Pistons are of conical form. H.P. piston is of cast iron, comparatively heavy for balancing; the I.P. and L.P. pistons are of cast steel. The piston rings are of cast iron. The H.P. and M.P. have each one ring with two packing grooves; where the ring is cut there is on the inside of each end of the ring a square shoulder ⅜ inch by 1 inch wide, over which is fitted a mortised-steel tie piece uniting the ends, thereby making the ring practically solid, but yet adjustable for wear. The L.P. have two rings, each in six sections, jointed as above.

All pistons are fitted with forged-steel followers, secured to the pistons by studs having square shoulders to prevent turning when they wear. They are ground true and burnished: each end is tapered, one for the reception of the piston and the other for the crosshead.

A thread is cut on the upper end of each piston for about an inch and a-half to receive a nut for use in handling the piston.

Piston rods are of annealed steel and tapered at each end to fit the piston or crossheads.

Crossheads.—Crossheads are of class "A" nickel-steel forgings, and carry a cast-steel slipper lined with white metal on the sliding face. The crosshead pins have tapered axial holes, the surface of the pins being flattened on each side, thereby reducing the bearing at the sides of the brasses and leaving a space for lubricant.

Crosshead Slippers.—The slippers are of cast steel, are secured to the crosshead by six 1½-inch tap bolts, and are lined with white metal.

Crosshead Guides.—The ahead guides are of cast iron, bolted at top and bottom to the strongbacks of the engine columns, and stiffened by webs on the back. The backing guides are of cast steel bolted to the ahead guides, with liners between to permit adjustment to wear.

Connecting rods are of the same material as the piston rods, and have axial holes, and forked at the upper end to carry the crosshead brasses, and the lower end is faced to receive the crank-pin brasses. The brasses are secured by a forged-steel cap and bolts.

The eccentrics are in two parts, are bolted together and keyed to raised seatings on the crank shaft, the ahead eccentrics being arranged for slight adjustment by means of a slot of greater width than the key, which permits their being shifted radially.

The eccentric straps are of composition lined with white metal. They are in halves, fitted with distance pieces and bolted together.

PROPELLERS.

There are two three-blade propellers; both are outward-turning for ahead motion.

The blades and the hub are of manganese-bronze. The hub is fitted on the tapered end of the shaft and held in place by longitudinal keys and a nut. The blades fit in recesses on the hub; the bolt holes are elongated to permit adjusting the pitch of the blades.

The blades, after casting, were ground and burnished, and the propellers swung between centers on a mandrel, and accurately balanced by removing metal from the heavier blades.

SHAFTING AND SHAFT BEARINGS.

The shafting is of class "A" forgings in six sections, viz: two sections of crank shaft, thrust, line, stern-tube and propeller shafts. The stern-tube section is coupled to the line shaft so as to permit withdrawal from stern tube. All shafting is hollow. The stern-tube shafts, which cannot be readily examined, are protected against the action of sea water by brass casing, shrunk on. Both the stern-tube and propeller shafts are fitted with heavy brass journals at the stern-tube and strut bearings.

All shaft couplings are of standard flange type, forged to the shafts, except the forward coupling of the stern-tube shaft and the outboard coupling connecting the stern-tube and propeller shafts. The former consists of a forged-steel collar secured to a raised seating on the shaft by keys, while the outboard coupling consists of a forged-steel sleeve, taper bored from each end to the center, for receiving the tapered shaft end. The sleeve is secured to the shaft end by four keys and two collars. The sleeve couplings are covered by a watertight cast-composition casing. the space between the casing and coupling being filled with tallow and rosin.

There are six main bearings for each crank shaft, two steady bearings for each thrust shaft, two spring bearings for each line shaft, two stern-tube bearings for each stern-tube shaft and one strut bearing for each propeller shaft.

There is one thrust bearing for main engine, of the ordinary horseshoe type, consisting of cast-iron pedestal bolted to cast-iron plate which is riveted to the foundation built in the hull structure. The lower part of the pedestal forms an oil trough in which the lower part of the thrust collars turn, the trough being fitted with a water-circulating coil for cooling the oil.

There are thirteen horeshoes of cast steel, made hollow for the circulation of cooling water, and lined with white metal on the bearing surface, each horseshoe being independently adjusted fore-and-aft by clamp nuts on screwed steel side.

The turning gear for each engine consists of a two-cylinder engine driving a worm gearing that meshes in the worm wheel keyed to the after crank-shaft coupling. There is in addition a ratchet attachment for turning by hand.

MAIN CONDENSERS.

There is one main condenser, located on the outboard side of each engine room; they are of horizontal, cylindrical, surface-condensing type. The shell is made of boiler plate; composition heads, and tube sheets of Muntz metal, the tubes and ferrules of Admiralty metal and are not tinned. The forward head of each condenser is fitted with a butterfly valve, so that the circulating water can be diverted overboard without going through the condenser tubes. Zinc plates are placed on the inside of each manhole plate. The shell is lagged with cowhair felt and covered with galvanized sheet steel.

Diameter of shell, feet and inches	6–07
Thickness of shell, inch	00 3/8
Length over all, feet and inches	15–04 1/2
Outside diameter, tubes, inch	00 5/8
Thickness	No. 16 B. W. G.
Length between the tube sheets, feet	13
Number tubes in condenser	4,838
Cooling surface, each, square feet	10,375

PUMPS.

Main Air Pumps.—A main air pump is provided for each main condenser. They are of the Fore River twin-cylinder, vertical-beam, single-acting type.

Diameter of steam cylinders (two), inches	12½
pump cylinders (two), inches	30
Stroke, inches	20

Main Circulating Pumps.—One centrifugal, double-inlet, circulating pump, driven by a compound engine, is furnished for each main condenser. Each pump has a capacity of 12,000 gallons of water per minute at about 265 revolutions. The pumps are arranged to draw from the sea, the main drain, and from the engine-room bilge, and discharge through the condenser or directly overboard, through the by-pass in the inlet and outlet condenser head, to the overboard delivery. The suction valves of these pumps are so interlocked that the sea-valve and the bilge or drain valves cannot be opened at the same time.

Diameter of steam cylinders, H.P., inches	10
L.P., inches	20
Stroke, inches	10
Diameter of pump runner, inches	42
inlet nozzle, inches	18½
outlet nozzle, inches	21

Feed Pumps.—There are four main feed pumps, of the Fore River, simplex, vertical, double-acting, piston plunger type, two in each engine room. They are arranged to draw water from the feed and filter tanks, and discharge to the main feed line direct or through the feed-water heater, and then into the main feed line.

Six auxiliary feed pumps, of the same type as the above, are located one in each fireroom. They have suctions from the feed and filter tanks, the sea, the boiler blowpipes, and the drainage manifolds in their own compartment, and discharge into the auxiliary feed line, overboard and into the fire main.

	Main Feed Pumps.	Aux. Feed Pumps.
Diameter of steam cylinger, inches	16	14
water cylinder, inches	9½	8
Stroke, inches	18	12

Feed Supply Pumps.—In each engine room there is one Fore River, simplex, vertical, double-acting, feed supply pump, having suction connections from the reserve-feed tanks in the

double bottoms, and the filling pipes from the ship's side. These pumps discharge into the feed and filter tanks and into the reserve-feed tanks.

Diameter of steam cylinder, inches	6
water cylinder, inches	6
Stroke, inches	8

Distiller Circulating Pump.—There is one distiller circulating pump in the starboard engine room, 12 inches by 14 inches by 12 inches, vertical, simplex, piston, Fore River type. This pump has a sea suction and discharges through the distillers overboard or into the flushing main.

Fire and Bilge Pumps.—One fire and bilge pump of the Fore River, simplex, vertical, double-acting type, is located in each engine room. These pumps draw water from the sea, the drainage system and from the bilges in their own compartment, and discharge into the fire main or overboard.

Diameter of steam cylinder, inches	12
water cylinder, inches	10
Stroke, inches	12

Feed and Filter Tanks.—There is located in each engine room one combined feed and filter tank of 5,500 gallons capacity. The filter box is built in the forward upper end of the tank. It consists of three separate compartments, each filled with the filtering material, through which the water from the main and auxiliary air pumps discharge passes successively before entering the feed tank.

Feed-Water Heaters.—A vertical, cylindrical feed-water heater is located in each engine room on the discharge side of the main feed pumps. The heating agency is exhaust steam from the auxiliary exhaust line, which enters the shell at the top, circulates around the tubes and drains at the bottom to the main or auxiliary condensers. The feed water passes through the tubes, from bottom to top. A slight back pressure is kept in the auxiliary exhaust lines, for properly heating the feed water, by adjustable spring-relief valves at the connections to the condensers.

Diameter of shell, inside, feet and inches	2–05¼
Length over all, feet and inches	10–04
Tubes, number in one heater	683
diameter, outside, inch	00⅝
thickness, No. 16 B.W.G., inch	0.065
length between tube sheets, feet and inches	7–05
Heating surface, each heater, square feet	829

Auxiliary Condensers.—There are four auxiliary condensers, one in each engine room, and one for each dynamo room. They are of the Fore River surface-condensing type, with combined air and circulating pump, of the Fore River horizontal, simplex type, attached to and under the condenser.

	Engine room.	*Dynamo room.*
Diameter of shell inside, feet and inches	2–01½	2–08⅛
Length over all, feet and inches	8–08½	9–00⅝
Tubes, number, each	458	782
diameter, outside, inch	00⅝	00⅝
thickness, No. 16 B. W. G., inch	0.065	0.065
length between tube sheets, feet and inches	7–04	7–07
Cooling surface, square feet, each	550.8	970.3

Auxiliary Air and Circulating Pumps.—

Diameter of steam cylinder, inches	8	10
air cylinder, inches	10	12
water cylinder, inches	10	12
Stroke, inches	12	12

Forced-Draft Blowers.—Twelve Sturtevant blowers, located two in each boiler compartment above the boilers, supply the forced draft. The blowers deliver directly into the firerooms, the closed-fireroom system being used. The engines are two-cylinder, upright and double.

Steam cylinders, diameter, inches	6
Stroke, inches	5
Fan, diameter, inches	66

ASH HOISTS.

For the removal of ashes the ventilator in each fireroom is fitted with angle guide strips to facilitate the hoisting of the ash buckets through same, and thence by trolley to the chutes at the ship's side. The hoisting is done by six Williamson

Brothers 4½ by 4½-inch reversible ash-hoist engines, located in the fireroom hatches.

DISTILLING APPARATUS.

The distilling apparatus is on the berth deck just forward of the engine hatches. It consists of three evaporators, of the horizontal straight-tube type, two vertical distillers, located in a trunk above the evaporators, and the necessary accessories. The designed capacity of the plant was 16,500 gallons of water per twenty-four hours, which requirement was most successfully met when tested.

	Evaporators.	*Distillers.*
Diameter of shell, internal, feet and inches	5–05	0–18
Length over all, feet and inches	7–06½	6–02¼
Tubes, number, each	30	223
diameter, outside, inches	02	00⅝
thickness, No. 12 and 16 B. W. G., inch	00.109	00.065
length between tube sheets, feet and inches	5–10½	4–06⅝
heating surface, square feet	275	165
Capacity each, gallons per 24 hours	5,500	8,250

TEST OF DISTILLING APPARATUS U. S. S. VERMONT, JAN. 4, 1907.

Time	Steam pressure, pounds.			Vapor pressure, pounds.			Distilled water gal's per 5 min.	Injection water, temp. of	Distilled water, temp. of
	Coil 1.	Coil 2.	Coil 3.	Coil 1.	Coil 2.	Coil 3.			
10·30	30	30	25	5	5	5	...	...	...
11·00	30	27	22	4	4	4.75	...	...	...
11·30	45	40	37	4.5	5	4.75	...	...	75
11·35	45	45	45	5	5	5	74	..	...
11·45	45	45	37	1.5	2	2.75	61	...	...
12·15	50	50	47	2	2.75	4.5	70	...	78
12·55	50	50	45	2	3	3.5	70	...	...
1·30	45	45	40	1.5	2	4	71	...	...
1·55	50	47	45	1	1.5	3.75	70	...	70
Av'g.	47.5	47.0	43.1	2.16	2.7	4.0	62.16	58.0	74.0

Gallons of water distilled per rate per 24 hours, 196.30.
Pounds of water distilled per square foot of H.S. per hour, 8.26.
Pounds of water distilled per square foot of C.S. per hour, 20.6.

NOTES.—
Designed capacity = 16,500 gallons per 24 hours.
Equivalent per hour = 687.5 gallons. (About 11.5 gallons per minute.)
Plant consists of three evaporators and two distillers.
Total H.S. = 825 square feet.
Total C.S. = 330.98 square feet.

REFRIGERATING APPARATUS.

U. S. S. "VERMONT"—TESTS OF 2-TON ALLEN DENSE-AIR ICE MACHINES NO. 199, NO 206.

Ice machine number.	Time.	Pressures, in pounds: Steam.	Pressures, in pounds: Compressor cylinder.	Pressures, in pounds: Expander cylinder.	Revolution per minute.	Temperatures, degrees Fahrenheit. Refrigerator rooms: Crew.	Refrigerator rooms: Junior and warrant officers.	Refrigerator rooms: Ward-room officers.	Refrigerator rooms: Admiral and Captain.	Refrigerator rooms: Air lock.	Sea water.	Atmosphere.	Scuttle-butt: Forward.	Scuttle-butt: Aft.
Ice Machine No. 199. January 24, 1907.	a. m. 8·00	96	...	...	110	53	50	58	54	56	36	56	36	36
	8·30	...	200	48	104	...	...	...	...	...	36	...	...	...
	9·00	120	225	65	108	...	...	...	...	...	36	60	34	35
	10·00	92	250	72	104	50	45	52	46	56	36	60	34	35
	10·30	105	250	72	116	...	...	...	...	...	36	61	...	...
	11·00	120	250	72	106	44	37	45	37	55	36	60	34	35
	11·30	122	250	72	114	...	...	...	...	...	36	62	...	...
	12·00	122	250	72	112	34	26	33	26	52	36	61	34	34
	p. m. 12·30	122	242	70	111	...	...	...	...	...	36	61	...	...
	1·00	122	260	75	107	22	19	23	20	49	36	63	34	34
	1·30	122	253	73	106	...	...	...	...	...	36	63	...	...
	2·00	122	255	73	111	12	13	23	13	46	36	56	34	34
	2·30	121	250	72	112	...	...	...	...	...	36	57	...	...
	3·00	122	247	70	114	14	9	14	10	45	36	58	34	34
	3·30	120	255	72	112	...	...	...	...	...	36	56	...	...
	4·00	120	255	72	112	10	8	11	7	43	36	65	33	33
Ice Machine No. 206. January 25, 1907.	a. m. 10·00	127	...	...	130	40	38	40	39	38	31	56	38	38
	10·30	122	95	23	120	...	...	...	...	...	31	56	...	...
	11·00	121	165	43	116	38	33	39	36	38	31	61	38	38
	11·30	122	220	62	108	...	...	...	...	...	31	60	...	...
	12·00	122	253	73	111	35	31	36	33	39	31	63	38	38
	p. m. 12·30	121	257	75	110	...	...	...	...	...	31	64	...	...
	1·00	122	255	75	108	30	25	30	27	37	31	65	37	36
	1·30	122	250	72	109	...	...	...	...	...	31	61	...	...
	2·00	122	250	72	110	25	19	24	21	36	31	63	34	33
	2·30	122	265	76	108	...	...	...	...	...	31	62	...	...
	3·00	122	255	74	112	12	11	15	13	34	31	63	34	33
	3·30	121	250	72	113	...	...	...	...	...	31	65	...	...
	4·00	120	250	73	112	12	7	9	8	32	31	65	34	33

Steam cylinder 9¾ inches. Compressor cylinder 7 inches. Expander cylinder 5¾ inches × 13 inches stroke. No. 199—3 cans ice at 11; ice tanks cut off at 11·12; 6 cans at 12; all cans at 2 P. M.

No. 206—3 cans ice at 1; all frozen at 2 P. M. (135 pounds).

NOTES.—Plant consists of two-ton ice machines and accessories. Each machine run independently on all circuits, beginning with atmospheric conditions throughout plant and continued until demonstrated that temperatures of refrigerating rooms and scuttlebutts were lowering uniformly and a full tank (135 pounds) of ice frozen.

This plant consists of two 2-ton horizontal Allen dense-air ice machines of the latest design and naval practice, supplying cold air to the cold-storage rooms, ice-making tank and the scuttlebutts. The official test of this plant showed highly satisfactory results.

GENERAL WORKSHOP.

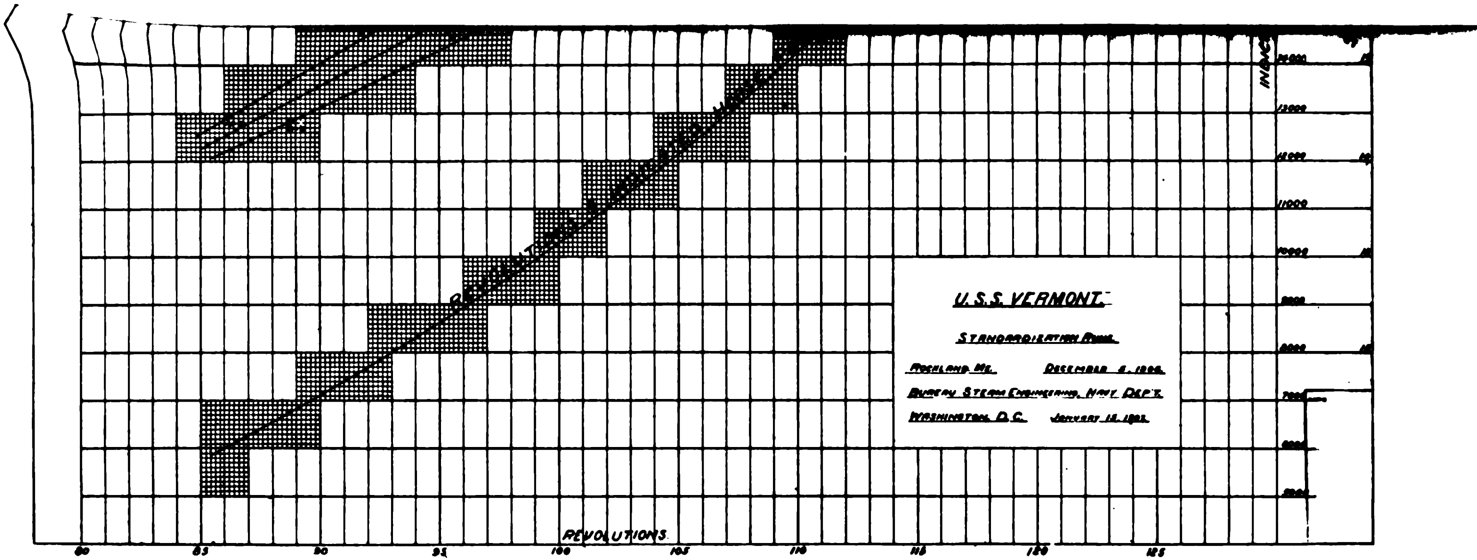
Located on the berth deck, between the engine hatches, is the general workshop. The tools are driven by a 10-horse-power motor. The following tools are installed: One 28 by 48-inch and one 14-inch lathe; one milling machine; one shaper; one 31-inch drill press; one 16-inch sensitive drill; one double-wheel emery grinder; one punch and shear; one grindstone; and bench vises.

An air compressor is installed in the port engine room for running pneumatic tools.

SCREW PROPELLERS.

The two propellers are of manganese-bronze and turn out-board when going ahead. They have three blades, and are designed as true screws. The surfaces of the blades are ground smooth with an emery wheel and edges filed sharp, the hub and blades are cast separate, the latter being bolted to the hubs by seven tap bolts each. The bolt holes in the palm of the blades are elliptical to permit of adjusting the pitch. After the pitch has been set the blades are secured against turning by composition chock pieces fitted in the space between the bolts and edges of the elliptical holes.

Diameter of propeller, feet and inches	17-09
hub, feet and inches	4-06
Pitch, feet and inches	18-00
ratio to diameter	1.014
variable between, feet	17 and 19
Helicoidal (developed) area, square feet	90.57
Projected area, square feet	77.88
Disc area, square feet	247.45



U.S.S. VERMONT
STANDARDIZATION RUNS
ROCKLAND ME. DECEMBER 6, 1906
BUREAU STEAM ENGINEERING, NAVY DEP'T.
WASHINGTON D.C. JANUARY 12, 1907
REVOLUTIONS
80
85
90
95
100
105
110
115
120
125
13000
12000
11000
10000
9000
8000
7000
6000
5000

BOILERS.

There are twelve Babcock & Wilcox water-tube boilers of the latest design, arranged two each in six communicating watertight compartments, with athwartship fireroom between each two boilers.

The boilers are identical with those installed in the *Louisiana* and described by Lieutenant R. K. Crank, U. S. N., in Volume XVIII, No. 1, of February, 1906.

OFFICIAL TRIALS.

STANDARDIZATION OF SCREWS.

The vessel was tried by the standardized screw method, progressive runs being made over the measured-mile course off Rockland, Maine, December 5, 1906.

From the data obtained on these runs the curves shown on Plate I were plotted.

The draught and corresponding displacement were as follows:

	Beginning.	End.
Draught forward, feet and inches,	24–02$\frac{7}{16}$	24–1$\frac{5}{8}$
aft, feet and inches, . .	24–10$\frac{9}{16}$	24–8$\frac{1}{8}$
Corresponding displacement, tons,	16,026	15,946

The results of the runs with and against the tide having been plotted as separate curves, the curve of true speed was obtained, from which it was deduced that a mean of both engines of 114.6 revolutions per minute would be required to give a true speed of 18 knots.

OFFICIAL FOUR-HOURS' TRIAL.

On the morning of December 7, 1906, the *Vermont* got under way and went to sea for the four-hours' trial. The weather was cold, squally and thick, with snow and rain. There was a moderate breeze from N.N.E. to a moderate gale from W.N.W. Sea moderate to rough.

Draught at beginning of trial, forward, ft. and in.,	24–1$\frac{3}{16}$
aft, feet and inches,	25–1$\frac{15}{16}$
Corresponding displacement, tons, ...	16,093
Average revolutions per minute for trial, . . .	117.12
Corresponding speed from curve, knots, . . .	18.33

The following is a synopsis of the data obtained:

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Mean steam pressure at boilers, pounds	264.2	
engines, pounds	259.0	258.0
H.P. steam-chest gauge, pounds	259.0	254.0
1st receiver (absolute), pounds	99.0	105.0
2d receiver (absolute), pounds	36.0	37.2
Vacuum in condensers, inches of mercury, mean	26.0	26.0

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Injection, degrees	45.0	45.0
Discharge, degrees	117.0	109.0
Hotwell, degrees	87.0	82.0
Feed water, degrees	177.0	158.0
Engine room, working platform, degrees	58.0	58.0
Firerooms, working level, degrees	50.9	50.9
Smoke stacks, average, degrees	450.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	117.16	117.05
Mean revolutions, both engines, per minute	117.10	
Pumps, main air	14.3	13.0
circulating	176.0	193.0
feed, d.s., per minute	22.0	24.0
fire and bilge	44.0	45.0
Dynamos	118.0	
Blower engines	138.0	
Speed of ship, in knots per hour	18.33	
Slip of propeller, in per cent. of its own speed, based on mean pitch	12.38	12.09
Air pressure in firerooms, in inches of water, mean	1.47	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	120.0	110.0
I.P. cylinder	45.6	48.6
F.L.P. cylinder	19.9	21.3
A.L.P. cylinder	19.8	21.3

	Starboard.	Port.
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	53.7	54.39

INDICATED HORSEPOWER.

	Starboard.	Port.
Main engines, H.P. cylinder	2,757.0	2,528.0
I.P. cylinder	2,833.0	3,016.0
F.L.P. cylinder	1,641.0	1,660.0
A.L.P. cylinder	1,633.0	1,756.0
total	8,864.0	8,960.0
Collective H.P. of both main engines	**17,624.0**	
Air pumps, main; Circulating pumps, main; Feed pumps, main; Hotwell pumps	358.0	
Other auxiliaries	267.0	
Total	18,249.0	

COAL.

Pounds, per hour, main and auxiliary engines, during trial...... 36,450.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	16.59
main engines, air, circulating and feed pumps, per square foot of grate surface	16.35
main engines, air, circulating and feed pumps, per square foot of heating surface	0.34
Pounds of coal per I.H.P. per hour, collective, main engines, air, circulating and feed pumps	2.03
Pounds of coal per I.H.P. per hour, all machinery in operation	1.942
square foot of grate surface, per hour	32.23
Cooling surface (main condenser), square feet per I.H.P. (main engine)	1.177
Heating surface, square feet per I.H.P. (total)	2.89

OFFICIAL TWENTY-FOUR-HOURS' TRIAL.

Immediately after the four-hours' trial the twenty-four-hours' endurance trial required by the contract commenced. The weather was very cold, thick and squally, with moderate to stiff gale from N.N.W. to W.N.W. Rough sea. The average indicated horsepower of the main engines on this trial was 14,935. Average revolutions, 110.34; corresponding speed from curve, 17.43. The following is a synopsis of the data obtained:

14

Steam Pressures. (Average of one-half hourly observations.)

	Starboard.	Port.
Mean steam pressure at boilers, pounds	236.2	
engines, pounds	234.7	234.9
H.P. steam chest gauge, pounds	222.2	219.8
1st receiver (absolute), pounds	82.6	92.0
2d receiver (absolute), pounds	29.9	32.0
Vacuum in condensers, inches of mercury, mean	25.9	26.5

Temperatures. (Average of one-half hourly observations.)

Injection, degrees	45.9	46.0
Discharge, degrees	111.2	116.5
Hotwell, degrees	82.0	86.4
Feed water, degrees	165.1	153.8
Engine room, working platform, degrees	64.2	63.3
Firerooms, working level, degrees	47.5	47.5
Smoke stacks, average, degrees	450.0	

Revolutions, or double strokes, per minute. (Average of one-half hourly observations.)

Average revolutions, main engines, per minute	110.4	110.2
Mean revolutions, both engines, per minute	110.3	
Pumps, main air	14.4	13.8
circulating	176.3	162.3
feed, d.s., per minute	19.7	22.4
fire and bilge	44.2	26.4
Blower engines	294.5	
Speed of ship, in knots per hour	17.43	
Slip of propeller, in per cent. of its own speed, based on mean pitch	11.21	11.26
Air pressure in firerooms, in inches of water, mean	1.04	

Mean Effective Pressures in Cylinders, in pounds per square inch. (Averages of cards taken at half-hourly periods.)

Main engines, H.P. cylinder	112.4	99.6
I.P. cylinder	39.2	42.7
F.L.P. cylinder	17.5	18.4
A.L.P. cylinder	16.9	18.4
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	47.73	48.37

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	2,440.0	2,152.0
I.P. cylinder	2,311.0	2,492.0
F.L.P. cylinder	1,363.0	1,427.0
A.L.P. cylinder	1,319.0	1,431.0
total	7,433.0	7,502.0

	Starboard.	Port.
Collective H.P. of both main engines	**14,935.0**	
Air pumps, main Circulating pumps, main Feed pumps, main Hotwell pumps	292.0	
Other auxiliaries	185.0	
Total, all machinery	15,412.0	

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	14.01
Cooling surface (main condenser), square feet per I.H.P. (main engine)	1.347
Heating surface, square feet per I.H.P. (total)	3.422

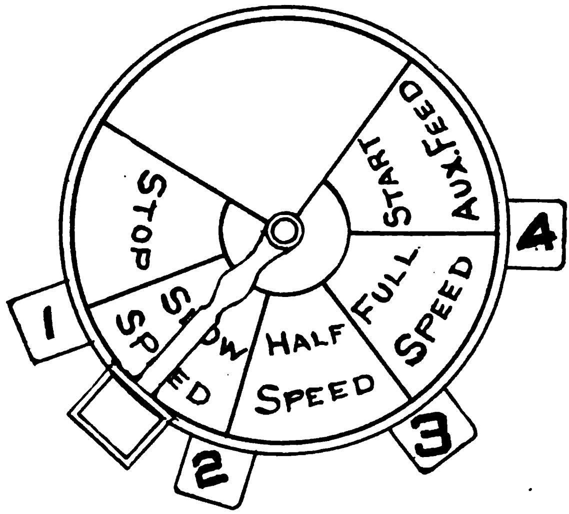
THE ADVANTAGES OF A SYSTEMATIC AND REGULAR METHOD OF WORKING THE FIRES IN A BOILER.

By Lieutenant Commander R. K. Crank, U. S. N., Member.

It is thought that the advantages of some regular systematic method of working the fires aboard ship over the old unmethodical way of "every man for himself" are so apparent to anyone, be he officer or enlisted man, who has had any experience with and without any systematic method, that it is unnecessary to dilate unduly upon these advantages. The average sailor-man will growl over the introduction of any innovation, for the reason that it is new and because he likes to growl; but, if one may judge from the experience on this ship (*Louisiana*), not a single fireman would vote to go back to the old way of working the fires.

The advantages of a regular, systematic method of working fires, by periodic signals to the firerooms, were deeply impressed on the writer on the trial trips of a number of the new and larger ships which he attended in the winter of 1905-1906, especially when the well-ordered methods in the firerooms of the *Louisiana* on her trial were contrasted with the comparatively confused conditions in the firerooms of the *Missouri* on the trial of that vessel by the same builders. These two examples alone were sufficiently convincing.

Assuming the advantages of a system to be indisputable, the writer submits these notes in the hope that they may contain some suggestion that may prove of use to officers on board those ships in which no automatic signals for working fires have been fitted or used and in which no regular system of firing has been employed.



U. S. S. "LOUISIANA."

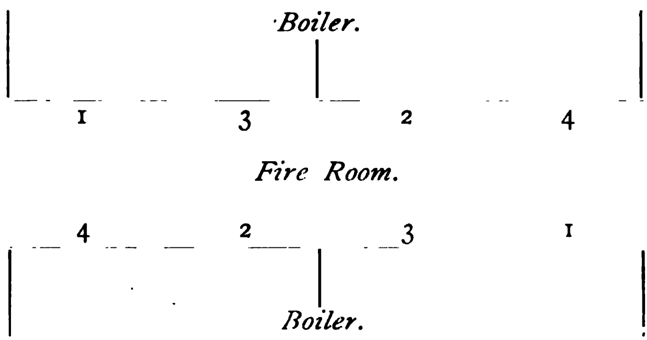
Sketch showing the method of fitting a fireroom telegraph for use as a transmitter of signals for working the fires in rotation and at fixed intervals. The figures by which the furnaces are designated are painted on pieces of sheet brass which are fastened opposite the lines between the divisions on the face of the dial. The pointer of the dial is fitted with an extension, cut so as to form a frame for each number and painted so as to indicate beyond possibility of mistake what number it is intended to signal.

To make any ordinary signal other than a signal for working a particular fire, the pointer would be placed in the middle of a division, as shown in the sketch, where the signal "Slow Speed" is indicated.

In August, 1906, about one month after the *Louisiana* quitted the Norfolk Navy Yard on her first service after commissioning, the fireroom telegraphs were fitted so as to permit them to be used as transmitters for sending firing signals. The accompanying sketch shows the manner in which each transmitter was fitted. As will be seen, the numbers by which the different furnaces or fires are designated were marked in large figures on pieces of sheet brass which were secured to the back of the dial and opposite the lines of demarcation between the divisions already on the face of the dial (that is, the divisions marked "Stop," "Slow Speed," "Half Speed," etc.). When it is desired to send a signal such as "Stop," "Slow

Speed," etc., the pointer of the telegraph is put in the middle of the proper division on the face of the dial (in the sketch, "Slow Speed" is indicated). When the telegraph is in use for transmitting signals for working the fires the pointer is placed opposite and in line with one of the figures by which a particular fire is designated. A man is stationed at the telegraph in the starboard engine room, with a time-piece, and instructed to send the signals in rotation at a stated interval between two successive signals. After trial it was found that, for ordinary cruising speed (12 to 13 knots), with about seven-twelfths boiler power, the interval between successive signals should be about two minutes. This interval, for any particular ship, can be determined only by trial.

Facing the boiler, the furnaces are numbered as shown below, viz:



This numbering of the furnaces is slightly different from the numbering at the time of the trial for reasons that will be given later.

The cycle and sequence of operations followed in working the fires are as follows, viz: Whenever a signal was received in the fireroom, the fire indicated by that signal (No. 1, 2, 3 or 4) was *raked,* and, immediately afterwards and without any further signal, the fire next to the one just raked and on the same side of the boiler with the one just raked, was coaled. Always, on receiving a signal, the first operation was to *rake* the fire indicated by the signal and then coal the one next to it. The sequence, therefore, would be:

Signal.	1st operation.	2d operation.
No. 1	Rake Fire No. 1	Coal Fire No. 3.
No. 2	Rake Fire No. 2	Coal Fire No. 4.
No. 3	Rake Fire No. 3	Coal Fire No. 1.
No. 4	Rake Fire No. 4	Coal Fire No. 2.

Taking any one fire (as No. 1), it will be seen that it is raked in one cycle and coaled in the next: also, referring to the diagram of the numbering of the furnaces, it will be seen that there is always a bright fire and a freshly-coaled fire in the same side of the boiler. With the numbering of the furnaces on the trial, there would be always two bright fires together in one half of the boiler and two green fires together in the other half of the boiler.

In order to counteract the persistent tendency of the firemen to fire too heavily and to pile the coal up in the front of the furnace, particularly in the cases of those men who had been accustomed to firing Scotch boilers by firing "on the dead plate" where the fresh charge was allowed to coke before being shoved back, orders were given that not more than four shovelfuls of coal should be put on any one fire at one time. With a regular system of firing it was possible to go into any fireroom at any time at sea, and, by an inspection of any fire, to determine whether or not the fireman had been carrying out orders and whether or not he was firing properly and efficiently. Although great improvement resulted from the use of the system and the limiting the amount to be put on any one fire, it has not been possible, in the five months of steaming since commissioning, to get the older firemen into the habit of carrying level, light fires, despite entreaties, prayers, profanity and dire threats. With a good system of firing signals it would be possible to make better firemen, for a particular type of boiler, from green men who had never fired at all, but who were intelligent and willing to do as they were told, than from old men who consider themselves past masters in the art of firing. An efficient feed-water regulator would prove of the greatest assistance, for the average water-tender is so constantly exercised and preoccupied, with a water-tube boiler, in watching the water level in the gauge glass and

tending the throttle of the feed pump or manipulating the check valve that he devotes little or no attention to the working of the fires. With a systematic method of firing and a feed regulator that would maintain a constant water level there would result a regularity of steam pressure and an evenness of distribution of the work amongst the boilers in use that would conduce greatly to economy of steam production, to the maintenance of a regular speed, to the life of the boilers and to the peace of mind and comfort of the entire engineering personnel. We hear of special courses of instruction for every special class of men in the service except those who have to do with the motive power. When the conditions of the service will permit, special instruction on special ships should be given in firing and the rudiments of engine driving and the care of as to firing guns the results would be little short of amazing; machinery. If as much thought were given to firing boilers but, lacking these Utopian conditions, much may be accomplished by sea-going officers with the material on board. The adoption of any sort of a system of firing, by the increased regularity and economy in the production of steam and the gauge it affords on the abilities of the men, will amply repay the slight effort needed to put it in operation.

A NEW FORM OF AUTOMATIC ASSISTANT CYLINDER FOR VALVE GEARS.

By Luther D. Lovekin, Associate.

The subject of assistant cylinders for relieving the forces of inertia and gravity in connection with reciprocating valve gears is one which the writer has given years of attention, and at this writing 115 of these cylinders are in practical use in various naval and merchant vessels throughout the world.

Several forms have been used in order to obtain the best results in each particular case, and, like all other devices which require a thorough scientific investigation being made in order to fully understand the principle upon which it is based, this cylinder has been no exception to the rule of conservatism so prevalent among designing engineers. This is not surprising, when we consider the number of various devices that are brought before the engineer daily for his consideration, and with only a limited amount of time at his disposal for reviewing the same.

Being fully cognizant of these conditions, I have contributed from time to time various articles dealing with this subject, so as to bring the entire matter before the engineering world gradually.

I believe the time has now arrived when I can safely state that all required conditions have been fully met by some particular form of assistant cylinder among my various designs.

Within the past six months, however, I have brought out a cylinder which embodies the chief characteristics sought for by the designer, viz: simplicity, economy, automatic regulation, ease of application, and an assurance of the theoretical requirements being fulfilled in practice as closely as possible.

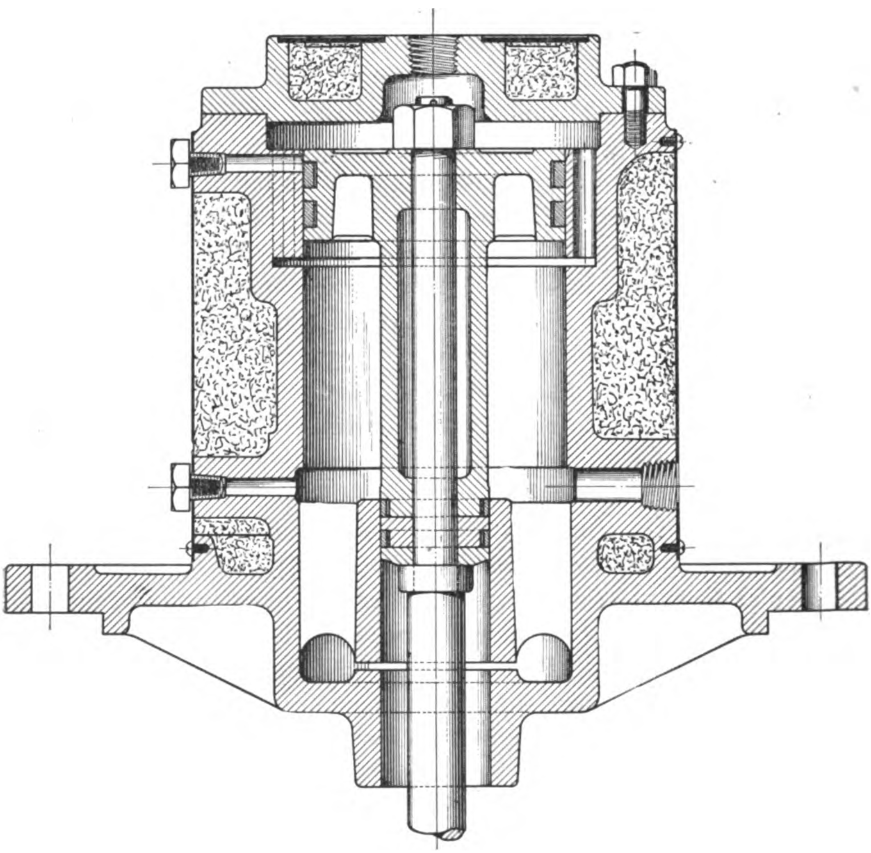
I take great pleasure in presenting herewith a full and com-

plete description of my automatic assistant cylinders, which I hope will be of interest to all concerned.

If a further investigation is required, I would refer to the different articles which have appeared at various times, and which I believe form an exhaustive treatise on this subject, viz: "Assistant Cylinders," published in the JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS," Volume XVI, Nos. 1 and 3; and Volume XVII, No. 3. Also "Proceedings of the Society of Naval Architects and Marine Engineers," November, 1902 (prize essay).

THE LOVEKIN IMPROVED ASSISTANT CYLINDER

TYPE "A"



THE LOVEKIN IMPROVED "ASSISTANT CYLINDER" FOR VALVE GEARS.

Type A.—This type comprises a valve-chest bonnet and "assistant cylinder" combined in one piece, and is used in connection with an engine having piston valves, which necessarily requires a circular valve-chest bonnet or cover over the same.

It will be noticed that the "assistant cylinder" receives its supply of steam from its own valve chest or receiver, and consequently requires no piping or valves whatever. It is entirely automatic in its action, and has the least number of parts possible to secure the desired results. It is designed in such a manner that the circular steam-inlet port at the lower end serves the purpose of admitting steam from its own receiver, and also forms an automatic drain for releasing any water that might collect in the lower part of the cylinder. The upper end of the "assistant cylinder" is designed to secure the best results in each individual case; the particulars of same are determined by the theoretical diagram (which decides whether no compression top is required, or otherwise). It is drained automatically, through the ports shown, into the lower part of the "assistant cylinder," and thence to the steam chest below.

NOTE.—Both upper and lower ends of cylinder are automatically drained during every revolution of the engine.

Piston.—The piston, as will be seen, is of the differential type; the diameter of the lower end being determined by the velocity of steam necessary at the inlet port. Both of these pistons have packing rings of the usual form, and of such width as is found necessary for riding over the narrow slotted ports in upper and lower cylinders, as shown.

The conditions of admission and compression are designed for each individual case in order to balance the forces due to inertia and gravity.

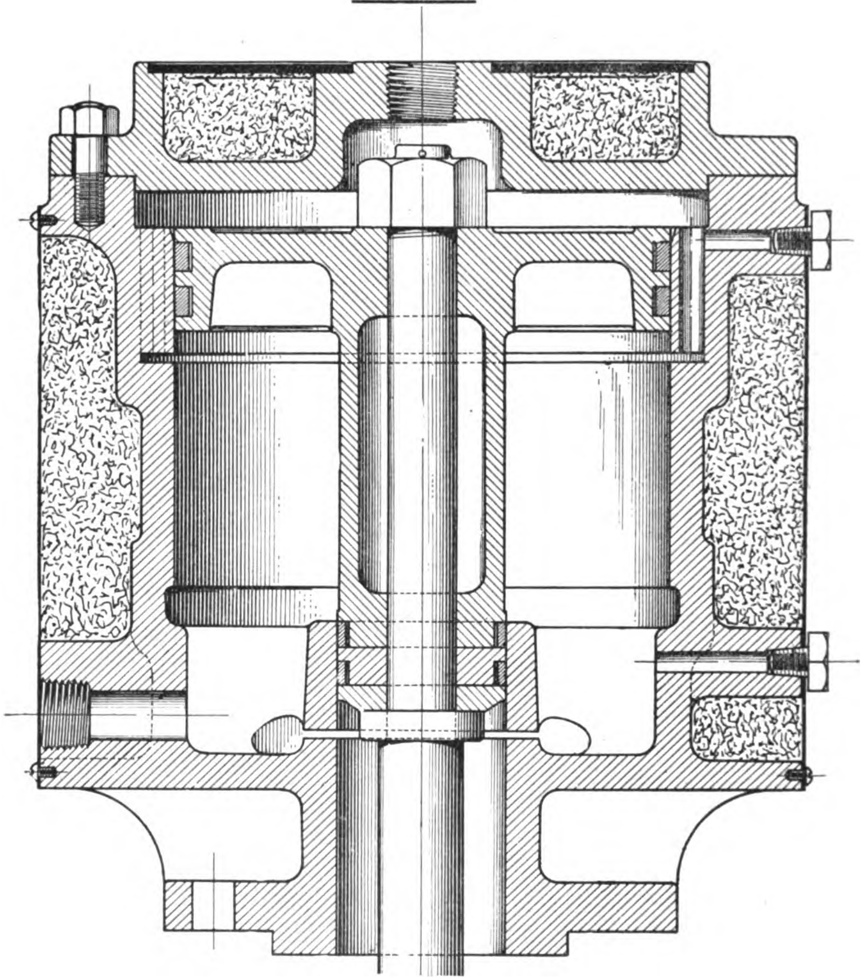
It will be noticed that there is *no exhaust port whatever;* consequently the only steam used in connection with a cylin-

der of this type is the amount necessary to make up the losses due to radiation and condensation.

The principal feature of this design is the ability to form a movable steam-tight packing in the lower cylinder and thus

THE LOVEKIN IMPROVED ASSISTANT CYLINDER

TYPE "B"



allow compression to be maintained in the upper cylinder. This is accomplished by means of packing rings in the piston,

as shown ; and necessitates *the lower cylinder bore being of a length equal, at least, to the travel of the main valve.*

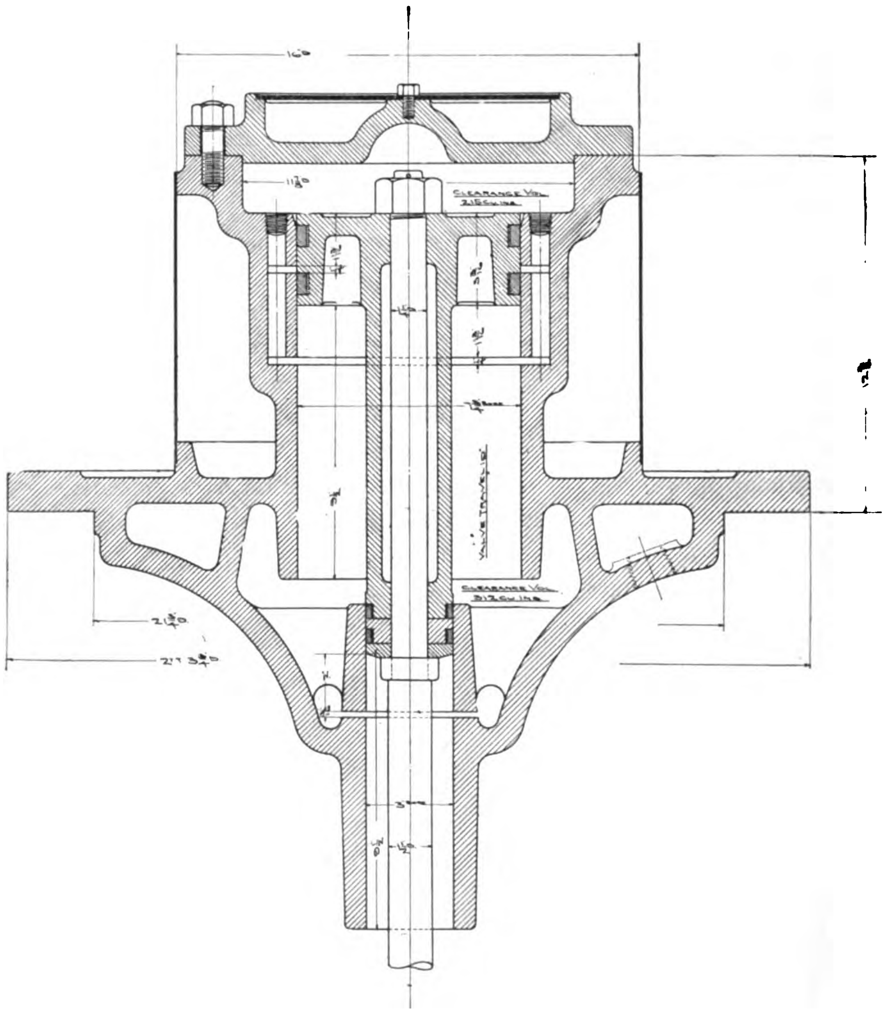
NOTE.—The description of assistant cylinder as given for Type A also serves for illustrating the action of Types B and E, as each of these as shown are fully automatic and require no pipes, no valves to manipulate, and have no complication of parts. They can also be fitted to any engine, old or new.

Type B.—Type B cylinder has the same characteristics as Type A.

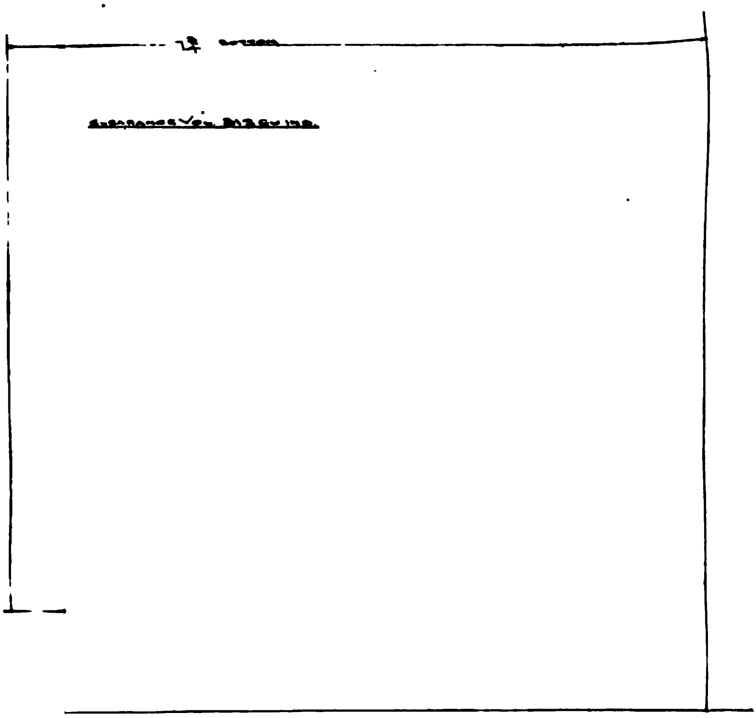
This cylinder is designed for use in connection with flat slide valves, and is either bolted directly to the steam-chest cover, or secured to the top of steam chest in the event of steam-chest covers being arranged at the end of the cylinder.

Type E.—This type is used in connection with high-speed engines, where the inertia top plays an important part. It can be designed to fulfil the same requirements as Type D. Each cylinder, however, is dependent upon its own receiver pressure, and consequently when fitted on the L.P. valve gears they will be of larger diameter than Type D.

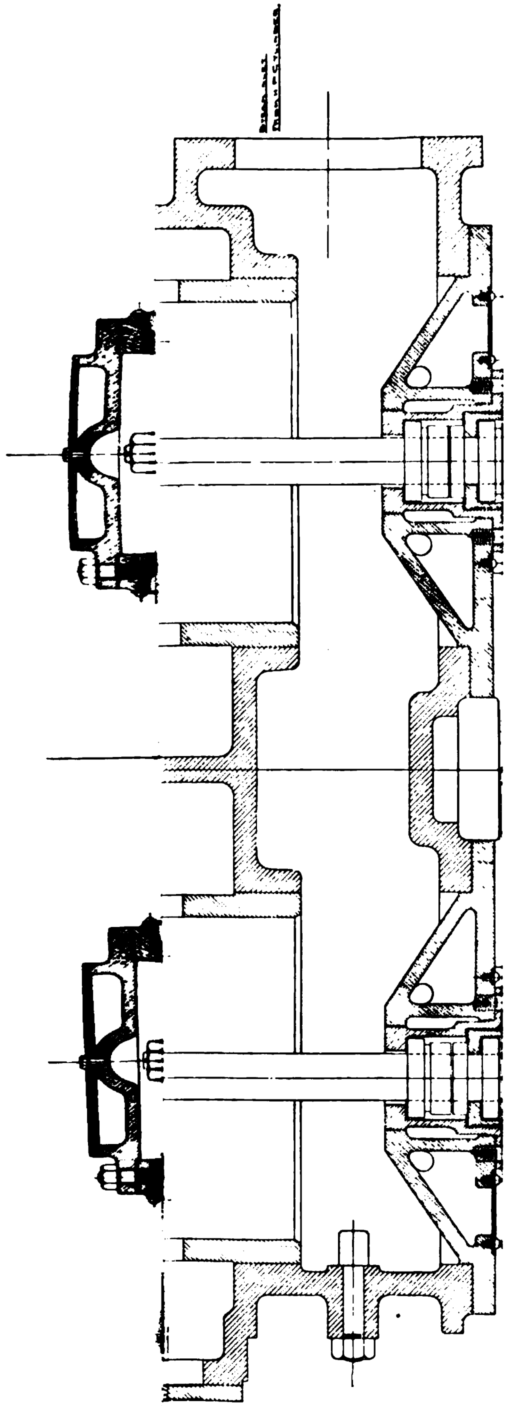
The chief characteristics are the same as described for Type A.



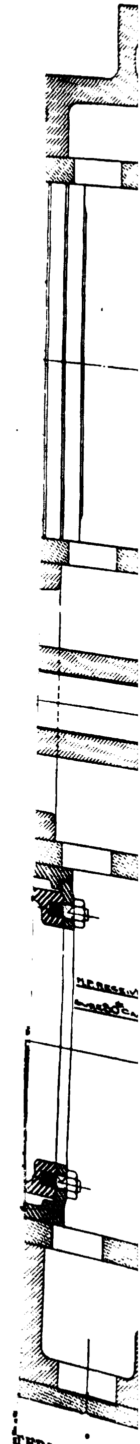
PRELIMINARY DESIGN OF TYPE E, FOR THE VALVE GEAR OF THE INTERMEDIATE CYLINDER, U. S. BATTLESHIP "MICHIGAN."



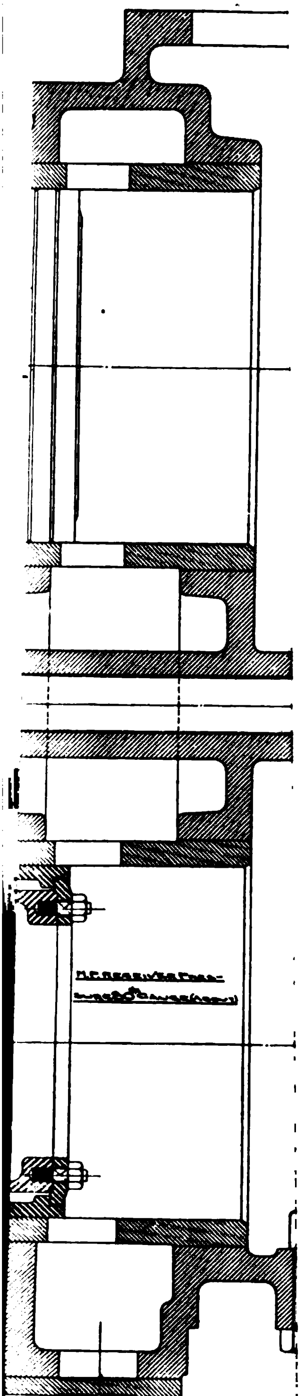
. BATTLESHIP "MICHIGAN."



. S. BATTLESHIP "MICHIGAN."



TERMEDIATE C



TERMEDIATE CYLINDER, U

NOTES.

SOME CHARACTERISTICS OF COAL AS AFFECTING PERFORMANCE OF STEAM BOILERS.

By W. L. Abbott.

For the purpose of determining, if possible, some simple rules by which the value of coal screenings could be determined without a continuous boiler test going on in a power house, the Chicago Edison Company proposed a series of tests. Mr. Bement was placed in charge of these, and the results and conclusions given are drawn from his reports.

In general these tests were made to determine the effect of varying percentages of ash and of different grades of fineness upon the value of the coal. To determine the effect of fineness or size, a large sample of coal, amounting to 100 tons or more, was taken and screened through sieves of different mesh until we got several piles of coal, each of approximately uniform size, and with each of these samples boiler tests were made. The results of these are shown in the diagram (Fig. 2), the upper curve indicating the horsepower obtained, or the *capacity*, and the lower curve the *efficiency* derived from these various sizes. It will be observed that the grade of coal nearest in size to a mesh ¾ inch square gave the highest capacity and at the same time the greatest efficiency. The reason for the variation in the values of the different sizes is partially explained by the fact that the smaller sizes carry a greater amount of ash, but it is not ash alone which accounts for all of the difference.

To further investigate the effect of size upon capacity and efficiency a series of 62 tests on ordinary screenings were analyzed, and the efficiencies and capacities which were obtained from these tests are plotted in diagrams (Figs. 4 and

5). A curious and unexpected result is shown. As might have been expected, screenings having an average size of 0.1 inch or under gave the lowest results. As the average

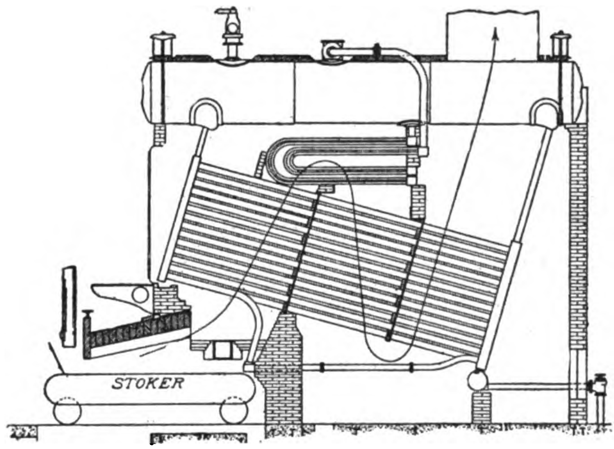


Fig. 1.—Elevation of Boiler and Stoker Employed in the Experiments.

size of the coal increased up to about 0.3 inch, the efficiency and the horsepower increased with it. From that point, as the coal became coarser, the efficiency dropped off until reaching a size something less than ½ inch, from which point it again increased.

Referring to Fig. 2, coal through a ¼ inch square screen produced only 108 horsepower, yet a size of fuel known in Illinois as No. 5 washed coal, which will pass through a ¼ inch round hole (a smaller aperture than the square opening) will produce as high as 600 horsepower under the same boiler.

It is probably true that if coal which contains so much dust that we would be unable to get one-half boiler capacity out of it, were put through a washer and freed from this objectionable fine dust only, not more than 3 per cent. of the combustible would be lost in the process. This would indicate that if the coal operator could free his coal from this 3 per cent. of dust, either by washing or, perhaps easier, by fanning,

he could vastly improve the quality of his coal at a very slight loss of 3 per cent. of the coal mined. In washing coal as much as 10 or 15 per cent. of combustible matter is sometimes removed, to get rid of less than that per cent. of ash. The coal could be improved in quality nearly as much by the simple expedient of removing 3 per cent. of dust, such as would go through a 20-mesh screen.

Another test, not directly bearing on the value of the fuel, was made to determine the efficiencies of fire beds of different thicknesses. As these tests were all made on chain-grate stokers, it was an easy matter to regulate the thickness of the fire to a nicety. The results obtained are shown in the diagram Fig. 6, the upper curves indicating the horsepower obtained with two different sizes of coal.

To determine the effect of ash upon the value of fuel we conducted a series of tests upon a large and uniform lot of coal. The results are shown in the diagram (Fig. 3). In this diagram the solid curve indicates horsepower and the curve with dashes indicates efficiency. The first test was made with coal containing something less than 10 per cent. of ash, and the successive tests with increasing amounts. When the ash content had been increased to 40 per cent. we could still burn the coal, and the coal would heat the water up to the boiling point, but it would not produce enough heat to make steam. Therefore, coal containing 40 per cent. of ash was absolutely valueless. This curve (Fig. 3) I consider the most interesting and satisfactory of any that we obtained during these tests. Those two curves—one of efficiency and one of capacity—are shown reproduced in Fig. 7, and the curve drawn through them which we have taken to indicate the value of the coal, because coal has one value in proportion to its efficiency and another in proportion to the boiler capacity that can be generated with it. Therefore, with a curve drawn through the other two, we may have the true indication of the commercial value of the coal. The scale on the left is drawn from 0 to 100, 100 being taken as a standard in coal having an ash of 12 per cent.

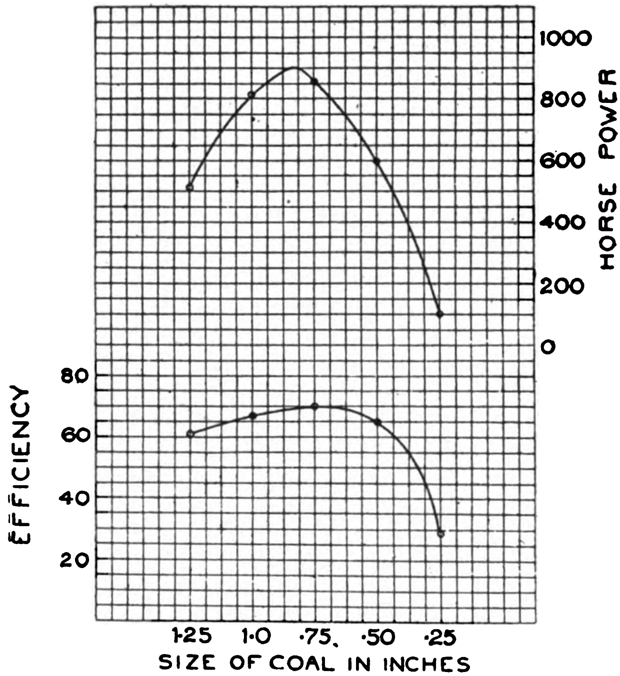


Fig. 2.—Effect Produced in Steam Generation by Coal of Varying Sizes.

The apparatus employed in the researches to be considered consisted of two Babcock & Wilcox boilers, one being fourteen tubes high and eighteen wide, of approximately 5,000 square feet of heating surface, fitted with a chain-grate stoker of 75 square feet in area, which discharged the gases of the fire from under an ignition arch 5 feet long, immediately among the tubes of the boiler. This boiler was also fitted with a Babcock & Wilcox superheater, having an approximate area of 1,000 square feet. The other apparatus employed in one of the series of tests differed only in sizes; its boiler was twelve tubes high and sixteen wide, contained 4,000 square feet of heating surface, provided with a superheater, and served with a chain-grate stoker of 66 square feet in area. Fig. 1 is a sectional elevation of the larger boiler.

INFLUENCE OF ASH IN COAL ON CAPACITY AND EFFICIENCY.

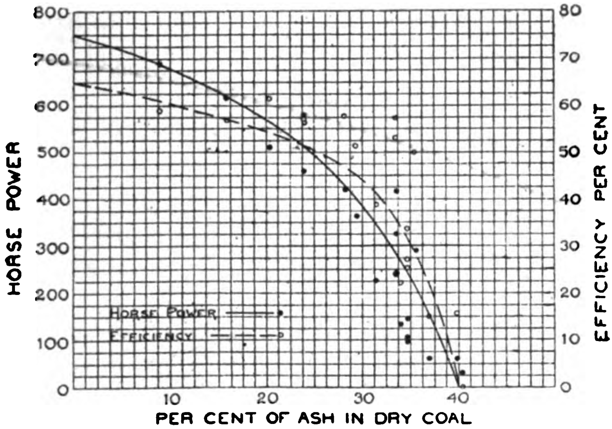


Fig. 3.—INFLUENCE OF VARYING PERCENTAGE OF ASH IN IN COAL.

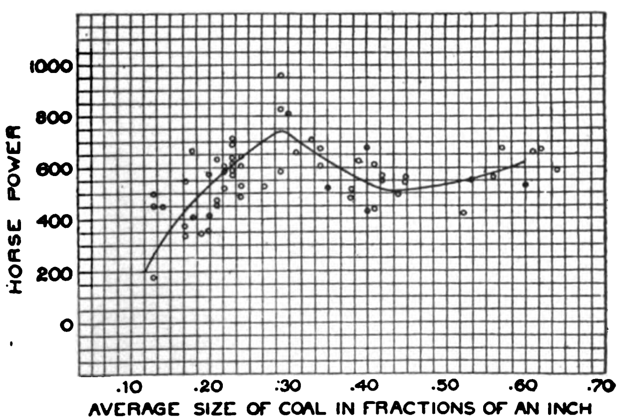


Fig. 4.—EFFECT OF SIZE IN COAL SCREENINGS ON CAPACITY PRODUCED.

Fig. 3 gives results of eighteen tests made to determine the effect of varying quantities of ash associated with coal.

Figs. 4 and 5 illustrate the result of sixty-two tests. With each the size of coal as measured by screens with square openings ranging in dimensions from 0.25 inch to 1.50 inch, advancing by 0.25 inch, and the average sizes of coal, as shown

at the base of the diagrams, were calculated from sizing tests made with these screens, and represent the dimension in fractions of an inch of openings in a screen which would allow one-half of the coal to pass through and the other half to go over the screen, and it is this that is designated as its average size.

Fig. 4 shows the effect produced on horsepower output, owing to this variation in size of the coal, and Fig. 5 illustrates the resulting efficiency from the same cause and for the same tests.

The curves of efficiency in this diagram illustrate a constant heat efficiency produced through the boiler for a full working range in thickness of fire, insuring not only maximum excess of air, but incomplete combustion loss as well, yet efficiency remained uniform, and the only opportunity for the "skilful" and "intelligent" fireman is in selecting that thickness best suited to capacity requirements.

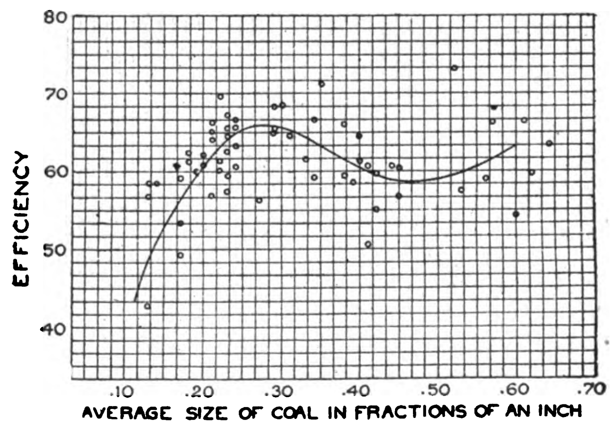


Fig. 5.—EFFECT OF SIZE OF COAL SCREENINGS ON EFFICIENCY PRODUCED.

The coal used in these two series of tests was very uniform in size and ash content, and for these reasons was well suited to the purpose of the experiments. In the series with thickness of fire, from 4.5-inch to 8.5-inch, what is known as No.

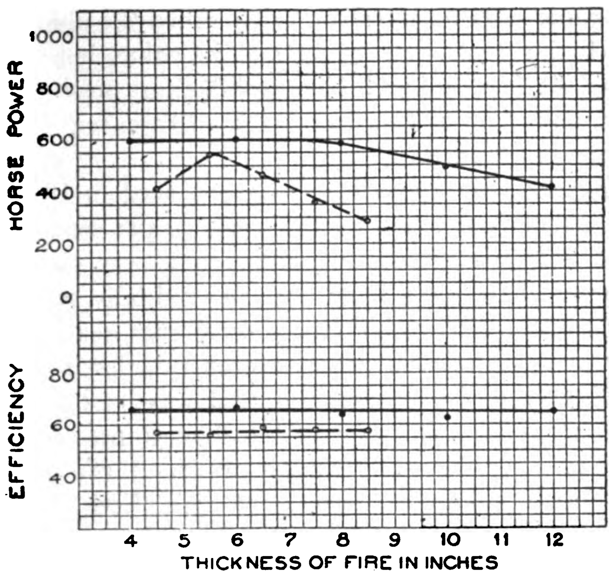


Fig. 6.—SHOWING UNIFORM HEAT EFFICIENCY WITH VARYING THICKNESS OF FIRE.

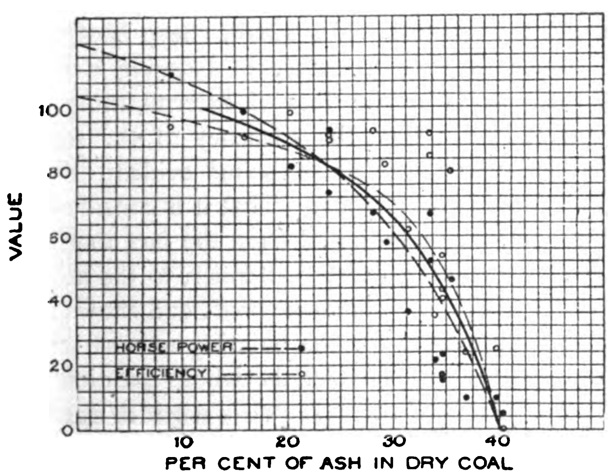


Fig. 7.

5 washed coal was used, a size which passes through a screen having round openings 0.25 inch diameter. With the other series and a larger range in thickness, washed screenings were employed.

This paper does not presume to lay down the ultimate laws by which fine coals may be graded in value, but rather to point out the fact that such laws, although at present obscure, do exist, and that our conclusions drawn from numerous tests are as herein indicated.

During the year 1905 there was produced in Illinois and Indiana about 50,000,000 tons of coal, 40 per cent. of which was 1¼-inch screenings, and, although it was not in every case separated from the lump, we can truthfully say that this 40 per cent., or 20,000,000 tons of screenings, was sold at the mine at an average price not to exceed two-thirds of its cost of production, and this same fine coal was used for making steam at an average efficiency of less than 50 per cent.

These two facts are sufficient warrant for further investigation of this little known subject.—"Page's Weekly."

THE POSITION OF THE SUBMARINE.

The building of very big warships enhances the value of very little warships; and that brings to the fore the policy of constructing submarines. If a small, cheap boat, like a submarine, stands as good a chance of bagging a two-million-pound *Dreadnought* as she does of exterminating a one-million-pound *Majestic,* then the efficiency of the small boat is distinctly advanced. So much being evident, the question remains: What is the chance? That it is considered to be an increasing one may be gathered from the fact that every first-class Power is devoting greater attention to the acquisition of a satisfactory design of submarine boat; and shows more readiness to provide money for the building of such craft. It is therefore opportune to consider the whole question in connection with the details of the past year's work in our own Navy and of the great advance lately attained, as a consequence of research work, in propelling machinery for submarine use. We have in commission, or in course of construction, over forty submarines; but in view of the activity of foreign Powers it is more and more to be regretted that our Admiralty reduced the number to be laid down during the current financial year from

twelve to eight, and the facts since disclosed strengthen the objection to this course which we expressed at the time.

Our chief rival in the possession of weapons for submarine warfare is France. The French Government were ahead of us in adding submarine boats to the navy of the Republic, and now holds the superior position so far as numbers are concerned. According to an Admiralty return, France had thirty-nine submarine boats completed last spring—as against our twenty-five—and fifty building, or projected, as against our twenty-three. Some of the French boats appear, on paper at any rate, far more powerful than our own. Thus the French had eighteen boats in construction, of a displacement of 391 tons, each fitted with no less than seven torpedo tubes; whilst there were six others under construction, each of 383 tons, and having six torpedo tubes. Our own boats of the "C" class, of which there are eleven building, are of 313 tons, and have but two tubes each, although they each carry four torpedoes. There is every probability, however, of a considerable increase in the size and power of the new British boats.

The British fleet of submarine boats has, however, the great advantage of being composed of homogeneous groups, whilst the French vessels are a very mixed collection. How far this would affect tactics in this entirely new field of warfare it is difficult to say, but it is probable that it would have a considerable influence. The original idea of operation with the submarine boat was that a single one would creep out and catch the battleship or big cruiser unawares. The same theory was held when the torpedo boat first came to the fore, but it was speedily found to be ill-founded; both maneuvers and the small amount of war experience that has accrued having shown that attack in group is the proper maneuver. Of course, if the torpedo boat could make reasonably sure of surprise and of the trustworthiness of its weapon, the single-unit attack would be the most profitable; but concealment until within torpedo range is by no means to be depended upon; and thus when attacking in groups a second or third may get its blow home while attention is being devoted to one or two others.

With the submarine the details are different, but the ultimate principle is much the same. The submarine has a means of concealment denied to the torpedo boat, but she is very much slower, and when she takes advantage of her means of concealment, by submergence, she cripples her power of attack enormously in some respects. It may be concluded that no belligerent vessels during any future war will lie at anchor, unless in a well-protected harbor, or in waters so far distant as to preclude the possibility of submarine attack; we have, therefore, only to consider maneuvers under steam. An American naval officer, who has made submarine vessels a study, has said that "the submarine, *when within torpedo range,* is superior to the battleship, since the battleship is vulnerable to the torpedo—the weapon of the submarine." This writer, Lieutenant Halligan, does well to emphasize the words he put in italics; but even when within torpedo range the submarine has not victory assured. Torpedoes fired from a submarine do not always hit their mark, although the application of the gyroscope has done wonders in adding to their trustworthiness. A British boat having four torpedoes on board, or a French boat with seven tubes—we are not aware how many torpedoes the latter vessels carry—might have a further chance; but unless the torpedo were to be fired at random, the boat would have to come to the surface again to sight for a second shot, supposing the ship had not, in the meantime, steamed out of range.

In order to make an effective attack upon a ship under way, it is necessary that the submarine boat should intercept the vessel in her course. The legend surface speed of the "C" class of British submarines is 13 knots, and their submerged speed for 3 hours is put down at 8½ knots. These are really remarkable rates of traveling for vessels handicapped in the manner submarine boats necessarily are, but it is anticipated that the "D" class will make 15 or 16 knots on the surface and 9 knots submerged. Some of the forthcoming French boats have been assigned 13 knots and 8 knots respectively for surface and submerged speeds, whilst the two most recently

projected vessels are intended to have 15 knots and 8 knots; but the designs are not yet complete.

We will suppose, however, that a submarine boat has reached the 14-knot and 9-knot standard. If the boat were to sight a hostile ship approaching directly towards her at 16 knots when at a distance of 4 miles, it would take a quarter of an hour for the ship to reach the submarine, supposing the latter to be stationary. That would be a piece of great good luck for the submarine, and, though it might occur, the probability is hardly great enough for it to be depended upon. We will suppose, however, that, in place of the boat being right in the enemy's path, the ship is steaming on a course that would pass the position occupied by the submarine at the time of sighting at a distance of 2 miles, the distance at which the ship was discovered being again 4 miles. The submarine would have her enemy as she sighted her 30 degrees before the beam. To take a simple case: if the boat, traveling at 9 knots, made directly for the course the ship was pursuing, it would take her rather more than 13 minutes to do the 2 miles which would bring her to a point cutting the ship's course, or, say, nearly 13 minutes to come within torpedo range. In the meantime the ship would have been occupying about the same time to cover the 3½ miles that would lie between her first sighted position and the point where the two would meet. Under these circumstances, the submarine would be within striking distance, and able to fire one torpedo.

The tactics are not necessarily those that would be pursued, but the original conditions supposed are favorable to the boat; that is to say, the luck would be in her favor. No time has been allowed for her filling her tanks to assume the submerged position, and she is supposed to have maintained a direct course. What part of the run would be made under water would be a matter for the discretion of the commander, who would be governed by the light and the conditions of weather; but it would be wise to keep the conning tower above the surface as long as possible, because it is difficult to keep an object in sight by the periscope, and also extremely difficult to judge

distance. The conning tower awash, however, makes the position of the boat visible by the wave it creates—a more noticeable thing than the tower itself in some states of the sea and atmosphere, when the boat is traveling at high speed.

In ordinary daylight the ship would be seen by the submarine at a greater distance than 8 miles, but it must be remembered that, even when the boat is highest out of water, the deck is elevated only a few inches above the surface. Standing on the top of the conning tower, a man would have a horizon distance of not much more than 3½ miles, whilst a height of 50 feet gives only slightly over an 8-mile radius. In daylight, however, there would be a greater chance of the submarine being sighted; and in that case the ship would alter her course so as to avoid the slow-moving submarine; for at present it is doubtful whether any effective means have been devised for a counter attack on these boats when they are below the surface. The most promising suggestion has been the use of small torpedoes with light charges, sufficient, however, to fatally damage so vulnerable a structure as a submarine boat. As they would deal with slow craft, these torpedoes need not be so fast as those used for swifter ships. They could be carried by boats and easily handled. Of course, there is the chance of going over, as well as under, the submarine, even if her position were known; but the torpedoes would be comparatively cheap, so that a fair number might be expended; and if the submarine had been sighted and were known to be attacking the ship, her position could be very fairly estimated, as her lack of speed does not give her much choice. The value of torpedo boats, destroyers, and pinnaces to act as vedettes for discovering submarines and attacking them with quick-firing guns, or with light torpedoes, will be understood. Even if the small craft could not injure the submarines, they might prevent them from coming up to take observations, and thus frustrate their object.

The improvements in the design, and consequent advance in efficiency, of the submarine boat has been a remarkable feature of recent naval history. The various naval authorities con-

cerned have, however, so well kept their secrets that there is not much to say on the subject. At the meeting of the American Society of Naval Architects and Marine Engineers, held in New York last month, an instructive paper on "The Development of Submarines" was read by Mr. L. Y. Spear, a member of that society. In this paper are given some interesting particulars of a trial of the submarine *Fulton*, as carried out by the United States Navy Department last June. The *Fulton* was of the *Adder* class, but has been somewhat altered and much improved in detail since she was built. The *Adder* is a single-screw boat, with the usual gasoline and electric means of propulsion. She is 64 feet long by 11 feet 9 inches beam, and displaces submerged 122 tons.

The program was for the *Fulton* to leave Newport Harbor and proceed at full speed in the light condition to a stakeboat anchored in the open sea. She was there to submerge and find a target 10 nautical miles distant to seaward. This target consisted of two ships' cutters 300 feet apart, and marked by a yacht 250 yards west of the target. All observations during this attack were to be made by the periscope. As it was feared torpedoes might be lost, the *Fulton* was simply required to expose her periscope within torpedo range, and then, submerging the periscope, to pass through the target. She was then to return submerged and pass round a stake boat three miles distant and again attack the target, this time making observations from the conning tower only, the eye-piece of the periscope being removed. She was then to return submerged over the course, pass to the light condition, and proceed under her main engines, charging batteries while under way. After charging batteries she was to remain submerged for 12 hours. The whole trial was to continue for at least 24 consecutive hours, during which the vessel and crew were to be entirely self-sustaining.

These were the conditions laid down, and the *Fulton* more than fulfilled them. She was submerged 15½ hours out of the 24; during the remaining 8½ hours she was cruising on the surface. During the 12 hours' continuous submergence the full crew and one observer were on board, and no fresh

air was supplied until the end of the test. The average depth maintained during the attack was 20 feet, which was sufficient to entirely submerge the periscope. During the 10-mile run to the target the periscope was momentarily exposed at intervals of about 2 miles; the boats, on account of their small elevation, were difficult to pick up. The final observation was made at a distance of 825 yards from the target.

These particulars of an interesting trial give a very fair account of what a submarine boat can accomplish. How the trials of our own craft would compare with the run of the *Fulton* we have no means of knowing. There is little doubt, however, that the submarines of the Royal Navy are second to none, and are probably in advance of all others.—"Engineering."

THE COALING OF WAR SHIPS.

Some coaling experiments of considerable value have been recently conducted by the Admiralty in the Mersey. His Majesty's ships *Vengeance* and *Cornwallis,* on the 28th of November, last, shipped between them about 1,900 tons of coal at Liverpool. The object of the test was to obtain reliable information concerning the facilities of this port for coaling operations, and it appears that the results obtained are satisfactory to the Admiralty. It was intended to ship 1,000 tons on either vessel, but the coal for the *Vengeance* being short, that vessel took on board only about 904 tons. This vessel was coaled by Clarke's patent automatic barges, one being employed on each side of the ship. The operation was completed in 4¾ hours, or at the rate of over 241 tons per hour, or about 121 tons per hour per barge.

The barges with which these satisfactory results were obtained are fitted with bucket elevators working along the bottom of the barge and up into small towers. The barge is provided with a false bottom. taking the form, therefore, of a large hopper, into which the coal is charged from the railway trucks at the wharf. In the bottom of this hopper are a number of slides, and on these being opened the coal falls into the buckets immediately beneath, which travel in the space

between the false bottom and the hull proper of the barge. The full buckets, containing about 5 cwt. of coal apiece, are weighed automatically, and the weight registered. The coal is then tipped into adjustable chutes leading to the bunker hatches or ports on the vessel to be coaled. The operation is practically continuous, and a special feature of the device is that the control of the coaling operations is practically in the hands of one man. The crew of the barge all told only number about six men, so that labor is practically reduced to a minimum.

When it is further added that the recent operations were handicapped by want of trimmers, and that, had sufficient labor of this class been forthcoming, the 904 tons would probably have been transshipped in about 3½ hours, it will be realized at once that this system of coaling is at once rapid and economical. The further trouble of obtaining some 400 or 500 men for ordinary coaling operations is obviated, and the possibility of delay due to lack of labor thus reduced to a minimum. The barges seem to go far towards fulfilling two of the most important conditions concerning the coaling of vessels of war—namely, readiness and prompt discharge.—"Engineering."

GERMAN REGULATIONS FOR GAS-ENGINE TESTS.

The German Associations of Engineers, engineering firms, and large gas-engine builders have issued a series of rules and regulations which are to govern in future the efficiency and output tests of gas engines and gas producers. These are given in detail in the "Zeitschrift des Vereines Deutscher Ingenieure," and their object is to render uniform the conditions under which tests of this class of installations are carried out, with a view to obtain results comparable with other results obtained on the same basis throughout the country. The regulations state the points on which the tests are to bear, both as regards producers and gas engines. The number and duration of the tests would vary according to their purpose, and they are to be planned and agreed upon beforehand, after tak-

ing into consideration the type of plant dealt with and its conditions of working; in the case of tests of special importance made with a view to the final taking over of a plant, or with reference to deductions or premiums, the interests involved are also to form a point for consideration. Tests for taking over a plant are to be made as soon as possible after the plant has been started working; the builders, however, will be given a stated time for private tests, and for effecting final improvements. The extent of this time and other conditions are to be fixed in the contract. For ascertaining the fuel consumption of gas producers the test has to last over eight consecutive hours without interruption. For establishing the consumption of liquid or gas fuel, with a constant load on the engine, tests of about one hour's duration are sufficient at the high loads; should results at low loads have to be ascertained also, tests of a lesser duration would be sufficient. In order to make sure that an engine is working under constant conditions, the temperature of the cooling water as it leaves the engine is to be measured at intervals. Once commenced, the tests have to continue without a break.

With a view to determine the efficiency of an explosion engine, tests of shorter duration under constant load are sufficient, but at least ten series of diagrams are to be taken. For investigations of particular importance, two tests following each other are to be carried out; and these will only be reckoned as valid when they have not been interrupted, and when the results of the two compared show only slight differences attributable to errors in readings. The average between the results of the two tests will be taken as final.

The allowances on output and consumption are to be fully agreed upon before the tests take place, either in the contract or when the test conditions are being drawn up. Should no special agreement have been arrived at on these points, the contract will be reckoned as fulfilled if the figure yielded by the tests is not more than 5 per cent. below the figure promised. But this allowance is only to apply to the maximum, and not to the regular normal output. The figure for the latter must under all circumstances be attained.

All indications with reference to pressure are to state whether absolute or not; temperatures are to be in centigrade degrees. For the mechanical heat equivalent the value 427 *m K g*=one calorie heat equivalent will be used, corresponding to one horsepower-hour = 632 calories.

The regulations enter into details with regard to the measures to be taken in carrying out the tests. The weighing of the coal and the taking of samples for analyses are also minutely stated. The reports showing the results of tests made are to give the type of plant, its working conditions, principal dimensions, the speed under various loads, when running light, and the variations in speed under a constant load, the method followed in recording the effective power, and so forth. The heat value of gases which do not contain heavy carburetted hydrogen can be calculated by formula should the determination by calorimeter not be practicable. The means for obtaining and recording volumetric measurements of the gases are also entered into.

The committee who have had charge of the preparation of the regulations in question include prominent German gas-engine builders and professors, among whom are Mr. E. Körting, Jr.; Mr. Richter, chief engineer of the Nuremberg Gas-Engine Works; Mr. Stein, Director of the Deutz Works; Dr. Stodola and Dr. Th. Peters, the latter Director of the Verein Deutscher Ingenieure.

All the clauses, their working, and their object, are fully explained by an appendix.—"Engineering."

GAS ENGINES AND MARINE PROPULSION.

From time to time, as advances are made in gas-engine practice, the question of the use of such engines for marine propulsion is raised. As one difficulty after another is removed by improvements or new systems, we reach a stage nearer the goal. It is evident that sure advances are being made in the right direction, but the difficulties still are many, and the steps somewhat small and slow, compared with the distance to be traversed. For instance, we take up a paper—plenty of

them have been submitted to societies and institutions—and read about the application of gas engines to ships. We hear of schemes for sets of engines capable of developing 3,600, 5,000 or more horsepower, and all sounds well; but we also know that the largest amount of power yet installed in the form of gas engines on board ship is only a mere fraction of these amounts. For this discrepancy there must be good reason, especially as the names of many able men are connected with the study of this problem.

The chief difficulties encountered in this matter arise from the necessity for any system of marine propulsion being adaptable to a large variety of circumstances, and, if the expression may be used, elastic in nature.

In the matter of fuel the system chosen must be suited to the supply obtainable in all parts of the world. In the matter of power it is necessary that there should be a wide range for working under all conditions of weather and circumstances; and in the matter of handling, the engines should be prompt and easy to reverse, and, in fact, perform all the duties at present carried out by steam. With regard to the first, it is futile to equip a vessel with a gas-producer plant using anthracite. The only fuel possible is bituminous coal, and that, too, in extremely variable qualities. But the producer must be prepared to use these different qualities with, at least, something approaching success. We are told that producers for bituminous coal have now been successfully designed, in which the various accessory appliances are either simplified or altogether abolished. Others have suggested the provision of coke-burning producers, the coke being made on board; this, however, usually being condemned as too cumbrous. It is by no means improbable that this producer difficulty, as well as others, will be surmounted. The subject is making rapid strides, and will no doubt develop to a degree which it is at present difficult to gauge.

Given a successful producer capable of using any grade of bituminous coal, there still remain such difficulties as reversing and reduction of power. This problem has been attacked in

several ways, but with results as yet not wholly satisfactory. Indeed, the choice at present rather falls upon that which is the least crude rather than that which is best. Reversing may be done by the propeller, but this only with small vessels; or it may be managed by rotating the cam shaft. Again, by adding considerably to the equipment it may be managed by electrical driving, or the result may be attained by the system suggested by Mr. J. T. Milton, M. Inst. C. E., in his recent paper before the Institution of Civil Engineers. Again, a suggestion has been put forth by Mr. A. Vennel Coster, in a paper before the Manchester Association of Engineers, that, with three sets of engines, in order to reduce power or bring a vessel to rest, one screw might be reversed while the other two still went ahead, and so on.

In the electrical drive, in which a gas engine drives a dynamo connected up to a motor on the propeller shaft, the horsepower of machinery to be paid for is three times that required to drive the vessel. In Mr. Milton's system of separate engine and compressor, with the example he gives of a 3,250 indicated horsepower engine, compressing gear requiring about 1,450 horsepower is necessary. Thus the machinery paid for amounts to 6,150 horsepower, while the net output of the installation is 3,250 horsepower. Of course, the propelling engine, being on the two-cycle system, would be relatively cheap. Machinery comprising eight or eleven cylinders, together with all moving parts, &c., does not impress one favorably. Mr. Milton's plan includes three expansion cylinders 25½ inches in diameter for the propelling engines. For the compressors two double-acting air cylinders 31 inches in diameter and two double-acting air cylinders 18 inches in diameter for the first and second stages of a two-stage compressor may be chosen, this being driven by a four-cylinder tandem gas engine, of cylinders 24½ inches in diameter. As an alternative, it is suggested to complete the second stage of the compression in one cylinder, 25 inches in diameter, instead of two of 18 inches, and to drive the compressor, a tandem two-cylinder engine, cylinders 34½ inches in diameter.

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Mr. Vennell Coster's system appears to us no less unsatisfactory. His proposal certainly involves no large auxiliary plant of compressors and driving engines; but to keep engines going full speed ahead and to counteract their effect by running other engines full speed astern seems a most unmechanical method of surmounting these difficulties. It is probable that the efficient reduction of mean pressure may be attained by fairly satisfactory methods, this being possible by Mr. Milton's plan, and, it would appear also, by certain other means. The chief difficulty, of course, in this is in the fact that the poor mixtures are difficult to ignite. With tandem vertical cylinders 50 per cent. reduction of power is possible by cutting out certain cylinders. Reduction of power below a certain point renders the engine liable to stop, no flywheels, or only very small ones, being possible in marine work.

A further difficulty which presents itself to the economical mind is the use to be made of gas from the producer when the engines are not required. On ceasing to draw gas for the purpose of running the engines, the temperature of the producers must be maintained by continuing the gas production. Something must be done with this. Mr. Coster uses it by running his engines at full speed, one counteracting the others. Mr. Milton makes no suggestion on this point, and we presume that that not required for driving his compressor plant would be allowed to run to waste.

Altogether, the problem is one full of knotty points. Although we are talking airily of installations of many thousands of horsepower, and are only building a few of a few hundred, and while the difficulties that suggest themselves at present appear, on present knowledge, difficult to surmount satisfactorily, we do not doubt that as progress is made, these will all disappear one after another. There is quite possibly a great future before this form of propulsion for marine purposes, but it would seem to be well to be not too ambitious till matters are a little more advanced. The two branches of the subject of producers and gas engines are, as yet, not far advanced even in land practice, and experiment in marine work

might well seem more advisable in schemes of moderate size than on the lines recommended by many recent writers on the subject, whose great aim appears to be mainly to obtain (on paper) some combination of cylinders capable of giving so many horsepower.—"Engineering."

A NEW ERA IN YACHT DESIGNS.

We have given this article the above somewhat comprehensive title, because the new measurement rule of the International Conference and the recent appearance of "Rules for the Building and Classification of Yachts," published by Lloyd's Register of British and Foreign Shipping, do actually promise that yacht designing will, in one department, assume a new and healthier character than it formerly did. Yachting has two characteristics: it is a sport, and it is a pastime. Under the former term are included all speed competitions; the latter describes cruising voyages. When sport and pastime can be enjoyed in the same vessel, the yachtsman gets the maximum amount of pleasure for his expenditure. It may further be stated that good sport may be obtained from vessels of honest design and wholesome scantling, even though they may take a minute or two longer over a given course than they would do if they were more flimsily built. For a long time flimsiness was the bane of yachting—that and lack of habitability. Committees and conferences have striven, by means of measurement rules, to bring shape within reasonable limits; but scantling has been too thorny a subject to be tackled, except in a very rudimentary manner in the case of small vessels. Glancing at the voluminous tables Lloyd's have now issued, one ceases to wonder at this reticence.

Our readers will remember that last January an International Conference on Yacht Measurement was held in London, for the purpose of arriving at an international rule of measurement which would give sufficient latitude to the designer to show his skill, yet would prevent the production of vessels which are simply racing machines. Whether the end will have been attained has yet to be proved; for the most knowing

of our yacht designers have become so skilful in the gentle art of tonnage cheating that there is no telling when they are cornered. The rule adopted is expressed by the equation:

$$(L + B + \tfrac{1}{2}G + 3d + \tfrac{1}{3}\sqrt{S} - F) \div 2 = \text{Rating in linear units,}$$

where L = length, B = beam, G = girth, d skin-girth—chain-girth (*i. e.*, girth difference), S = sail area, and F = freeboard; the units being either feet or meters. It will be seen that length, breadth and girth, together with sail area, form the taxable elements of the rule. To secure a habitable roomy vessel d is introduced, as it will have the effect of handicapping excessive hollowness of bilge. The minus sign before F indicates a premium on freeboard, though not a considerable one by any means.

There is one other important feature in the rule. The length L has formerly been measured on the water line, a fact which is largely accountable for the exaggerated overhang characteristic of modern racing yachts. No yachtsman, we think, except, perhaps, a few unballasted enthusiasts, will deny that these extravagant shelving ends are of the nature of excrescences. Anyone who has thrashed to windward in a latter-day racer, wind against tide, will bear witness to that. The old "plank-on-edge" yachts—the "lead-mines" of the Thames rule that held sway for so long—had their faults; but they would lie-to easily and with safety, and did not pound in a head sea like the big drum of a brass band.

A vessel with a long counter and a far-outreaching bow with considerable flare had this advantage as a tonnage cheater, that the water line was short when she was upright, but as soon as she heeled to the breeze the overhanging side became immersed, and added greatly to the virtual length of the hull, besides placing the center of buoyancy more to leeward. For these reasons one of the chief aims of the yacht designer was to get as much overhang as possible. This was carried to an extent undesirable for vessels designed to sail in the open sea, although less objectionable in smooth water.

It will be evident that without a considerable flare, or in-

crease of beam above water, the chief advantage of overhang would be lost; and the problem, therefore, was to tax flare to an extent that would check its abnormal development. It was some time before any means to this end, without unduly crippling design, was produced; but ultimately an admirable suggestion of a Danish naval architect, Mr. Benzon, was adopted at the Conference. His plan was to measure the girth of hull from deck to deck at the water line ending, to deduct from that twice the freeboard (a vertical measurement) at the same station, and to add the result (the difference between skin measurement and twice freeboard) to the water-line length. So far as can be judged by forecast, the principles involved in these two factors, the girth difference *d* and the handicapping of overhang at the water-line endings, should lead to a more wholesome and shipshape form of hull; whilst the inclusion of girth will tend to put a check on the exaggerated fin keel that appears as a deformity when a vessel is seen out of water, and which is a source of danger if she touch the ground. It must be acknowledged, however, that the fin enables lead to be carried low and gives quickness in stays.

The factors in the rule that we have named, including the tax on sail area, not only tend to produce a more efficient vessel in nearly everything except, perhaps, in mere speed, but they also incline to economy; the exception to the latter condition being the allowance for the factor *F*, freeboard, which, it will be seen, has the minus sign before it. Very few things, however, are better worth paying for than freeboard. In the Yacht Racing Association rule, to which we shall make reference later, freeboard is not taken into account, but other factors are common to both rules.

There remains one element needed in the composition of a yacht that should be a judicious compromise between an extreme speed craft and a comfortable cruising vessel; something neither a racing machine nor yet a kind of sea-going houseboat. The element referred to is strength of structure. Hitherto, whatever may have been attempted in the way of measurement rule towards producing a more desirable vessel

has been, as we have intimated, largely neutralized by the craze for flimsiness. We by no means wish to convey by this expression that hulls have been carelessly put together, or that materials have been bad; indeed, the very reverse is true. The trouble has been that builders, in their efforts to secure a low center of gravity, have aimed at putting as much of the total weight of hull as possible in the lead keel; and to this end have stinted material in the top sides and decks. The result has been an extremely expensive hull structure; for the cost saved in weight of material has been a trifle compared to the expense entailed by additional quality, and the extremely careful work needed. Yet, even with all this care and skill the yacht was a very delicate structure, not fit for the ordinary work of a sea-going vessel; whilst some of the least favorable examples were only capable of lasting through a race by being tuned up each time, and by a liberal use of the pump. Such vessels were of little value except as prize winners, and if they failed in this they became completely valueless. Even of the best it might be conceded that their mission was ended as soon as they became outclassed, as they speedily did, by the coming of a more extreme product of the art of measurement cheating.

These conditions tended to make yacht racing more and more a pursuit confined to the very rich, and therefore a decadent sport. Those in such a position as would formerly have enabled them to own and race big vessels were reduced to smaller craft, whilst many of the former small-craft men were driven out of the field. More vessels than one, after a couple of seasons or so, were unable to find purchasers for cruising at any price, and were broken up. A partial remedy for this state of things was the introduction of one-design classes, a number of boats being constructed from a single specification, which had to be strictly adhered to. The system has led to some good sport among the smaller classes, but it has its restrictions. In the first place, owners like to see in their craft characteristics that are not shared by every other vessel. This naturally is more apparent amongst those who

have knowledge enabling them to influence the design; but, in any case, a good yachtsman, by the fact of owning a vessel, has engendered in him something almost akin to parental affection, such, we believe, as has no parallel in the relations between man and any other inanimate object. It is this that accounts greatly for enthusiasm in a sport attended by no small measure of discomfort.

There is strong hope, as we have said, that the new International Regulations will do something towards putting an end to the defects which have been a burden on yachting; and that we may have a return to those more specious days when owners both lived and raced on their yachts; so that a boat, though built for racing, may still be worth something, even if she cannot fly a string of flags at the end of the season. When the scantling question was brought forward at the International Conference of 1906, it was seen that it was too full of complicated detail to be thrashed out by any congress. It was, however, determined broadly that scantling restrictions were absolutely necessary; and by a unanimous vote it was decided that Lloyd's Register should be invited to confer with other classification societies with a view to coming to an "agreement on a uniform rule for the scantling classification of sailing yachts."

Lloyd's accepted the duty thus suggested to them, and the result is the tables to which we have made reference, and which, it is safe to say, nearly every yacht designer has been studying closely of late. The tables, which are in two sets, one for metric and the other for English measurements, give the requirements for wood, steel, and composite yachts for each of the international rating classes from 5 meters (16.4 feet) to 23 meters (75.4 feet) inclusive. As is well known. for nearly thirty years past, Lloyd's have undertaken the survey and classification of yachts, but this has been chiefly confined to the larger craft, especially steam vessels. To have a little boat of 16½ feet built under survey, which will be compulsory if she is to compete in international races, is a new feature, and certainly one that has its advantages. As is

stated, "the constitution of Lloyd's is peculiarly well adapted for the administration of the new rules. Experienced surveyors are stationed in all the yacht-building ports, whose duty it will be, not only to see that the rules are complied with, but also to advise and assist the builders generally in carrying out the new requirements; and very complete instructions will be issued to the surveyors in order to ensure uniformity of practice in the different ports." The regulation, however, has been subjected to much adverse criticism, as it robs the designer of initiative; and there appears to be a general feeling that Lloyd's surveyors will be somewhat too heavy-handed for yacht work.

The scantlings alone occupy seven sheets, some of them containing over fifty columns; whilst the explanatory matter extends over 27 pages. It will hardly be expected we can give even a *précis* of the matter.* Wood, composite, and steel yachts are treated respectively in separate tables.

We have referred to the Yacht-Racing Association rule, and it will be of interest to add a few words about some former methods adopted for handicapping yachts for racing purposes. In the earlier part of the last century yachts were classed by the common tonnage rule, $L \times B \times D \div 96$. Length was taken on the keel, and this led to abnormal rake of stern posts —the earliest record of tonnage cheating—so the Thames clubs adopted a deck measurement, and in 1854 the celebrated Thames measurement was introduced. The formula was $(L—B) \times B \times \frac{1}{2} B \div 94$. Under this rule the great yacht races of this country were sailed for a period of nearly thirty years; and to meet it, until the rule was pressed to an undue extent, many fine yachts were built. The heavy tax on beam, and absolute freedom as to depth, ultimately led to the plank-on-edge model, in which sail-carrying power was entirely dependent on a deep lead keel. The old "lead mines" were, however, very weatherly, and could be depended upon to thrash out from the proverbial "N.E. corner against a S.W. gale," a consideration that was exceedingly comforting towards the

* For the convenience of our readers, we may state that the rules may be obtained by writing to the Secretary of Lloyd's Register of Shipping, 71 Fenchurch Street, the price being 5s. per copy.

end of the season. They were also absolutely safe from capsizing. It was, however, sometimes hard work to hold on on deck; and a race in the smaller classes—say one of the 3-tonners of the 'seventies—might remind one of a frolic on the back of a porpoise.

In 1880 the Yacht-Racing Association brought in a new rule: $(L + B)^2 \times B \div 1{,}730$; beam being again heavily taxed. In 1896 we find a more complex formula, in which girth was introduced. This had been suggested many years previously by Mr. Landseer Mackenzie, a successful amateur designer and Corinthian yachtsman. Sail area was also an element.

The '96 rule was superseded five years later by the existing Y. R. A. rule:—

$$\frac{L + B + \frac{3}{4}G + 4d + \frac{1}{2}\sqrt{s}}{2.1}.$$

In this formula we see introduced for the first time the factor *d*, girth difference, and there are—with the exception of freeboard—the same elements as in the international rule, though differently treated. The important principle included in the modification in the value of *L*, length, introduced at Mr. Benzon's suggestion, is, of course, absent in the older Y. R. A. rule.

Considerations of space prevent us following this interesting question further. We can only refer our readers to Mr. R. E. Froude's able paper, printed in the last volume of the Transactions of the Institution of Naval Architects, and to Lloyd's publications. The "Table of Tax Figures," in an appendix to Mr. Froude's paper, is especially instructive. Its purpose is to assess the tax on various elements, bringing them to a common denomination, and thus to enable a designer to determine to what extent a change in any given element can be balanced by changes in other elements. We give some of the figures below:—

......	*L.*	*B.*	*G.*	$\sqrt{s}$.	*d.*	*F.*
Existing Y. R. A. rule	0.44	0.12	0.13	0.27	0.035	...
Conference rule	0.54	0.14	0.11	0.22	0.032	— 0.038

Whether yacht builders will accept with a good grace the restrictions as to scantling to be put upon them by the action of the International Congress is a matter that remains to be seen; and their attitude will largely influence an extension of the legislation. As stated, there has been a good deal of adverse criticism. In some of the smaller classes the scantling is certainly generous in its dimensions, especially in the light of what some of the most successful designers of these craft, such as Mr. Linton Hope, have accustomed us to. However, the smaller classes are not so much affected by international agreements, and there will be plenty of racing outside Lloyd's rules. Whether, or to what extent, the clubs will fall into line with the suggestions now put forward remains also to be seen; but, in any case, the departure is one of great interest, and may prove the beginning of a better era for an uncontaminated form of sport, which, perhaps more than any other, fosters the best characteristics of our race.—"Engineering."

BRITISH NAVAL GUNNERY.

A report issued by the British Admiralty, concerning heavy gun practice in the Navy for 1906, gives some interesting figures, and shows a high standard of proficiency. The average points per man have risen from 68.2 last year to 80 this year, and forty-two ships out of eighty-eight that fired were above this average, while no less than fifty-eight were above the average of last year.

The first fifteen ships in order of merit made over one hundred points, and their firing was as follows: The *Drake,* flagship of Prince Louis of Battenberg, 146 hits out of 167 rounds; the *King Edward VII,* flagship of Sir William May, 130 hits out of 148 rounds; the *Cumberland,* 95 hits out of 113 rounds; the *Formidable,* 109 hits out of 130 rounds; the *Hindustan,* 121 hits out of 140 rounds; the *Carnarvon,* 78 hits out of 85 rounds; the *Duke of Edinburgh,* 115 hits out of 132 rounds; the *Exmouth,* flagship of Sir Arthur Wilson, 102 hits out of 138 rounds; the *Venus,* 78 hits out of 96 rounds; the *Hampshire,* 74 hits out of 90 rounds; the *Diana,* 76 hits out

of 99 rounds; the *Majestic,* 100 hits out of 126 rounds; the *Shearwater,* 39 hits out of 49 rounds; the *King Alfred,* flagship of Sir Arthur Moore, 120 hits out of 141 rounds; the *Good Hope,* flagship of Sir Richard Poore, 119 hits out of 147 rounds. Of these fifteen ships, the first in order of merit made 124.49 points and the last 101.44 points.

The *Bulwark,* the flagship of Lord Charles Beresford, made 93 hits out of 118 rounds fired. The second cruiser squadron, under command of Rear Admiral Prince Louis, takes first place in order of merit by squadrons. The score for this squadron averaged 98.7 points per man. The Mediterranean fleet came second, with the *Formidable* as the best ship, the average points per man being 93.5. The third cruiser squadron is third in order of merit, the *Carnarvon* being the best ship, and the points per man 90.8. The Atlantic fleet stands fourth, with the *King Edward VII* leading, and the squadron score at 88.5 points per man.

In the individual shooting some remarkable records were made. Seaman Baker, of the *Drake,* scored eleven hits out of eleven rounds with a six-inch gun in one minute. Scores of eight hits out of eight rounds and seven hits out of seven rounds were frequent with the six-inch gun. Petty Officer Sullivan, of the *Duke of Edinburgh,* made ten hits from ten rounds in one minute and a half with a 9.2-inch gun, and eight hits out of eight rounds with this gun appear to have been common. A petty officer of the *New Zealand,* with a twelve-inch gun, scored nine hits in ten rounds in two and three-quarter minutes, and a marine in the *Bulwark* made ten hits out of twelve rounds.

The Admiralty, in issuing the report, notes its extreme satisfaction at the very marked improvement in the results as compared with former years, including even those of 1905, when the shooting showed so great an advance over previous results. It further notes that this improvement is due to a higher standard generally throughout the fleet.

The report shows that in ten years the percentage of hits per gun per minute has increased with the 12-inch gun, from

.09 to .81; with the 9.2 inch, from .17 to 2.84; with the 6-inch, from .89 to 5.68, and with the 4.7-inch and 4-inch, from 1.83 to 4.98. Every ship in commission took part in target practice in 1906.—"Army and Navy Journal."

HISTORY OF THE TURBINE.

Now that the *Dreadnought's* steam trials have proved such a conspicuous success, it is interesting to trace the progress of the Parsons steam turbine from its first trial on the water nearly a decade ago. It is not proposed in this article to deal with turbine installations in vessels other than warships. It will hardly be disputed that the turbine, in its present stage of development, offers much greater advantages to men-of-war than to merchantmen, since there is a vast quantity of mercantile tonnage which the use of turbine engines would not at all benefit. Most readers of the "Naval and Military Record" will remember the interest that was excited by the remarkable speed of Mr. Parsons' experimental boat *Turbinia* at the Diamond Jubilee review at Spithead in 1897. This craft was something like a miniature destroyer in appearance, but could scarcely be classed as a warship.

The first vessel designed for war purposes to be fitted with turbine engines was the British Destroyer *Viper,* launched by Messrs. Hawthorn, Leslie & Co., on the Tyne, in 1899. On her trials she developed power equal to about 10,000 indicated horsepower, with which she attained the remarkable mean speed of 36 knots. The highest speed touched during the trials was 37.13 knots—a record which has not even yet been beaten. The displacement of the *Viper* was 325 tons, her length 210 feet, and beam 21 feet. She was followed by the *Cobra,* launched in 1900 by Messrs. Armstrong, Whitworth & Co. The *Cobra* was similar in most respects to the *Viper,* but 13 feet longer of 30 tons greater displacement. Though she did not quite reach the *Viper's* extraordinary speed, she proved herself faster than any other vessel afloat. Both these destroyers met with an untimely fate. The *Viper* was wrecked

off the Channel Islands during the maneuvers of 1901, her destruction being complete, though, fortunately, it was unattended with loss of life. The same was, unhappily, not the case with the *Cobra,* which was mysteriously lost in the North Sea, within sight of the Lincolnshire coast, while on her way from her builders to the Medway. She disappeared suddenly while steaming through a moderate sea, and no entirely satisfactory explanation of the catastrophe has ever been offered. Conjectures as to the cause ranged from a boiler explosion to collision with floating wreckage, but the theory most generally accepted was that she broke her back through structural weakness; and much ill-informed agitation ensued, both in Parliament and the press. It is to be doubted if this explanation was given much credence in the service.

The most serious result of these two disasters was the temporary set-back which was given to turbine propulsion. The opponents of the turbine were many, and included in their ranks not only those interested in the retention of reciprocating engines, but the professional obstructionists who oppose every new thing as a matter of principle. Had these two vessels, or even one of them, continued afloat and in service, there is no doubt their success would have led to the speedy general adoption of the turbine in the British navy, despite opposition; but the absence of sufficient data as to the value of turbine engines for continuous work at sea caused hesitation and delay in following up the matter. When progress was resumed it was in a more or less tentative manner, and continued so until the advent of the present first sea lord.

The next turbine-engined craft to take the water (again from the Hawthorn-Leslie yard) was the destroyer *Velox,* in 1902. Her dimensions were the same as those of the *Viper,* but the displacement went up to 400 tons. Her indicated horsepower was less, only 7,000, but she managed to reach an extreme speed of 33 knots with it. Her continuous speed in ordinary weather was 27.12 knots.

In 1903 Messrs. Hawthorn, Leslie & Co. turned out another turbine destroyer, the *Eden.* She differed considerably

from her predecessors, being among the first of the now well-known River type. Her dimensions are 220 feet by 23.5 feet. The horsepower developed was about the same as in the *Velox,* but the displacement being 550 tons, the trial speed was less, the maximum registered being 26.3 knots (mean). This was nearly a knot in excess of the contract speed. In heavy weather her design and larger size give her an advantage over the *Velox* in this respect.

So far the Admiralty had been content to confine the turbine principle to destroyers, but they now took a decided step forward by ordering turbines to be fitted to one of the four new cruisers of the Gem class, so that comparisons might be made between sister ships fitted with turbine and reciprocating engines. The dimensions of these four cruisers are 360 feet length by 40 feet beam; mean draught, 14 feet 6 inches, and displacement, 3,000 tons. They were launched in the latter part of 1903 and early in 1904, and trials were run early in 1905. The results of the four-hours' full-power trial are given below:

Amethyst, built by Armstrong & Co., 13,000 indicated horsepower; 23.63 knots, mean speed; 9 tons coal consumption per hour. *Diamond,* built by Cammell & Co., 9,868 indicated horespower; 22.1 knots, mean speed; 10 tons coal consumption per hour. *Sapphire,* built by Palmers; 10,200 indicated horsepower; 22.43 knots, mean speed; 11½ tons coal consumption per hour. *Topaze,* built by Cammell & Co.; 10,000 indicated horsepower; 22.25 knots, mean speed; 11 tons coal consumption per hour.

From this it will be seen that the *Amethyst,* fitted with turbines, is more than a knot faster than the best of her sisters, fitted with reciprocating engines, and this, too, at a coal consumption from 1 to 2½ tons less. Subsequent sea service has more than confirmed this result.

These facts probably had a large share in influencing the Admiralty to adopt the bold course (characteristic of Sir John Fisher's régime) of equipping the *Dreadnought*—a vessel six times the size of the *Amethyst* and vastly her superior in

fighting power—with turbine engines. How wise this measure was has been proved by the results of the *Dreadnought's* trials, and there is every reason to hope that the completion of the three great armored cruisers of the *Invincible* class, and of the several fast ocean-going destroyers now in hand, will witness a still greater triumph of the turbine.—"Army and Navy Military Record."

FOREIGN NOTES.

As proof of how rapidly war vessels deteriorate, mention may be made of the cruiser *Katoomba,* which has recently been stricken from the British naval list, although she is only sixteen years old, and was supposed to be a fine vessel when she was launched in the last of the 80's. A vessel of 2,575 tons, with a speed of 19 knots, she was very fast for her day; but the naval authorities have come to the conclusion that it was false economy to expend any money in repairing her, and she will be sold for what she will bring. Two years ago a vessel of the same class was repaired at an expense of $46,000. It was then found that she did not come up to modern standards, and she was put on the market and sold, bringing $10,000 less than her repair bill. In another instance, almost $100,000 was expended in repairing a British war vessel, that amount being about a fifth of her original cost, and inside of three years the ship was sold for $27,000, or $70,000 less than had been paid out to refit her. It is evident that the British Admiralty has come to the conclusion that it is far more economical to sell an out-of-date ship for what she will bring than to send good money after bad in a hopeless attempt to bring her up to date.—"Army and Navy Register."

THE "EXPRESS" COAL-BAGGING LIGHTER FOR COALING WAR VESSELS IN HARBOR.

The British naval authorities are experimenting at the Devonport dockyard with a new type of coal-bagging lighter for coaling war vessels when berthed or anchored in harbor.

The purpose of the invention is to provide means for filling bags with coal on board the lighter without any recourse to shoveling, and the automatic transportation of these loaded coal bags directly from the lighter to the bunkers of the warship without any handling whatever at any intermediate points. Rapidity in loading has been the object on the part of the designers, and with this appliance there is assured a coaling capacity of 60 tons an hour from each of the two transporters with which the lighter is equipped.

The hull of the lighter resembles that generally adopted for this class of craft. It is constructed entirely of steel and can carry a maximum load of 1,000 tons of coal. It measures 145 feet in length by 36 feet beam and 19⅛ feet molded depth, and has a draught of 14 feet when fully loaded. The hull is subdivided by means of three transverse bulkheads into four main compartments. In the forward compartment is accommodation for the crew; that next aft contains the boiler and steam-raising plant for driving the hoisting mechanism, electric-light installation, etc.; the two center compartments contain the coal. In the center of the craft, at the bottom, is a small reserved space where the hoisting engines, pumping engines, condenser and electric-light plant are placed. Above this area are the air-filtering and ventilating fan chambers. On the deck itself are two vertical towers or elevators fore and aft respectively for conveying the coal from the loading compartments to the warship alongside, while in the center of the deck are two slewing cranes for transshipping the coal from a collier to the lighter itself.

At the bottom of the lighter, extending practically its entire length on each side, and parallel to one another, are two galleries or filling rooms. These are about 7 feet in height, with a sloping crown at either side. On each side of this gangway are ranged benches at a sufficient height from the floor to enable the mouth of the sack when hung up to be just level with them. The crown of the roof slopes over these benches and the coal contained in the compartments above falls by gravity through orifices onto the benches and is raked by the men

into the open mouths of the sacks. Along the edges of the benches where the coal bags are suspended are fitted bag holders which hold the mouths of the bags open to their fullest extent while the men are raking in the coal. In order to prevent the coal from falling onto the floor of the filling rooms, fixed and portable screens are provided.

As rapidly as the bags are filled they are mechanically lifted onto an overhead rail along which they travel to the foot of one of the vertical elevators by means of a reciprocating pawl device. These elevators extend to a height of nearly 45 feet above the deck of the lighter and are constructed of steel with a crow's nest at the top from which the operator can easily follow and control the conveying operations upon the deck of the warship. Hinged to each elevator is a radial transporter arm long enough to reach over the deck of the vessel alongside. This arm has a vertical travel of 30 feet up and down the elevator, so it can be easily adjusted to the most suitable height; furthermore, it is so arranged that it can be turned through a considerable angle. When out of use the arm is packed vertically up the side of the elevator tower, completely out of the way. Each elevator is fitted with a Mackrow-Cameron "Express" transporter capable of lifting 120 tons of coal in bags per hour.

At the top and the bottom of each elevator tower is a grooved wheel over which travels an endless chain provided at intervals with hooks. The bag of coal, upon reaching the base of the elevator, is caught up by one of these hooks and lifted off the overhead rail extending through the galleries and is immediately hoisted by the traveling chain up the interior of the tower until it reaches a predetermined point, where the radial arm projects. It is then automatically released from the elevator chain and directed onto the radial arm, along which it is run to the point on the deck of the warship where it is to be delivered. Trunks are provided for returning the empty bags as rapidly as their contents are discharged to the filling galleries, and the cycle of operations is repeated until coaling is completed.

The two slewing cranes are each of $2\frac{1}{2}$ tons capacity and are of the high-speed Cameron type. They have a maximum overhang of 80 feet from the center line of the lighter, which enables them to reach well over the deck of the collier barge or other craft with ease. Their working radius is any distance between 8 feet and 40 feet, between which limits the loads can be lifted and dumped. In the case of a collier with the coal loose in the hold to be discharged into the holds of the lighter itself, there is a grab of one ton capacity, though they can be utilized equally well in transporting bags of coal from a barge to the lighter or *vice versa* as required, and can if the exigencies so demand assist in the coaling of the vessel already being served by the elevators. The coal drawn from the lighter is shot into the hoppers of the lighter through large hatchways in the deck. Thus while the lighter is coaling the warship alongside, the slewing cranes can be simultaneously replenishing the hoppers of the lighter itself from the opposite side. These slewing cranes are each capable of handling from 50 to 60 tons of coal an hour.

Naturally, while coaling operations are in progress the air within the bag-filling galleries becomes heavily impregnated with coal dust. In order to insure a perfect supply of clean, fresh air within this area the dust-laden atmosphere is withdrawn from the galleries by the ventilating fans, passed through the filtering medium, and fresh air supplied in its stead.

The boat is lighted throughout by electricity, there being some sixty incandescent lamps distributed through the loading galleries, engine and boiler rooms, and the crew's quarters. In order to facilitate coaling operations at night large arc lamps are fitted over the crow's nests on the elevator towers, and communication between the various parts of the lighter is afforded by electric bells and speaking tubes.

It will be seen that the transportation of the filled and the empty bags is entirely automatic throughout, the human element entering only in regard to the filling of the bags.

The bags of coal, upon reaching the warship's deck, are dumped down and wheeled away upon trolleys to the bunkers

by the coaling crew. The designers recommend, however, that two ports be provided in the sides of the battleships through which the radial transporter arms of the elevators can extend and connect with a system of overhead runners fitted on board and attached to the skid beams so as to form a continuous bar. In this manner the bags of coal would travel from the leading rooms on the lighter direct to the bunkers and the contents be there discharged, by which arrangement intermediate handling would be entirely obviated and coaling considerably facilitated and expedited. Furthermore, the decks would be left quite clear, and any necessity of clearing away ship's boats and gear, as is now usually the case when coaling is carried out on war vessels, would be dispensed with.

In the official trials recently carried out by the British Admiralty at Devonport for the purposes of testing the possibilities of this craft both in the coaling of war vessels and also the charging of the lighter itself from barges and colliers moored alongside, eminent success was obtained. In order to test the apparatus to the utmost these trials were extended over a period of four months, and in each trial a new crew for operating the lighter was employed, so that it was impossible for exceptional results to be obtained owing to the men becoming expert with the gear and consequently more expert in its manipulation as a result of continual practice. In the first place, the elimination of the preparations heretofore incidental to the coaling of a warship, such as the removal of boats and davits, etc., was emphasized, since in no instance was it found necessary to disturb the vessel's equipment in any way, the transportation bar being projected through any opening in the ship's side capable of admitting a 2-hundred-weight bag. As the coaling crew became expert in the removal of the bags of coal from the outer end of the transporter speed in coaling was accelerated, and the facility and lighter labor involved in the task was rendered very apparent. The most noticeable feature was the speed with which the vessel could be coaled by this system as compared with the other methods in vogue in

the dockyard, the capacity of the transporter far exceeding that attainable with the other processes.

In the course of the trials seven vessels of varying types were coaled, the quantity taken on board ranging from 1,000 tons for the *Duke of Edinburgh* to 172 tons for the *Monmouth*. In the case of the former vessel the trial extended over six hours, during which time 609 tons were placed on board, the remaining 391 tons being taken on after the trial. The highest coaling speed was attained in shipping 705 tons on board the *Isis*, when 41.75 tons an hour were placed on board from each transporter. In the final trial, in coaling the *Victorious* with 720 tons, the task was accomplished in 5 hours 40 minutes actual working time. Coaling was effected entirely by the transporters themselves without any assistance from the independent cranes. Had the crew been fresh, the work would have been completed in shorter time; furthermore, the men had had but little experience in handling the apparatus. Trials were also made with the cranes for transshipping coal from barges to the lighter itself. On this occasion 580 tons were lifted onto the lighter in 807 trips, the average load each trip being 14.37 hundredweight, or approximately 50 tons an hour. In the official tests of the capacity of the grab 14.84 hundredweight was the average of forty trips.—"Scientific American."

SHIPS.

ENGLAND.

The first-class Torpedo-boat No. 11 (formerly called the coastal destroyer *Mayfly*), constructed for the Admiralty by Yarrow & Co., of Poplar and Glasgow, had her official high-speed trial on March 6, when a speed of 27 knots was obtained during a run of eight hours, the contract speed being 26 knots. The consumption of oil during the eight hours was found to be 21.45 tons—*i. e.*, 2.68 tons per hour. A subsequent trial of 24 hours' duration was made at a speed of 12.11 knots, when the consumption was proved to be 8.79 tons for the 24 hours—*i. e.*, 0.366 ton per hour and 67.65 pounds per knot. On both these trials oil fuel alone was used. Although oil fuel has been tried repeatedly, both in this country and abroad, the Admiralty system is the only one that up to now has proved itself thoroughly successful.

New Destroyers.—The Admiralty have placed contracts for the construction of two new ocean-going destroyers, to be named *Saracen* and *Amazon,* which will be larger and more powerfully equipped than the five 33-knot ocean vessels at present building for the Royal Navy. The *Saracen* and *Amazon* are to have a displacement of 893 tons and 888 tons respectively. The *Saracen* is to be built at Cowes, by Messrs. J. S. White & Co., and the *Amazon* at Woolston, by Messrs. Thorncroft & Co. The *Saracen* will have a length of 272 feet and the *Amazon* a length of 280 feet.

The *Saracen* and *Amazon* are each to be equipped with two 4-inch breech-loading guns—the other ocean destroyers being armed with three 12-pounder guns—and will be fitted with engines of 15,500 horsepower, propelling them at a speed of thirty-three knots per hour. They will burn oil fuel, and are to be completed for delivery in 1908-9.

PERFORMANCES OF NEW BRITISH WARSHIPS DURING THE YEAR 1906.

Name of vessel.	Type of ship.	Builders of ship.	Makers of machinery.	Displacement.	Type of boilers.	Heating surface.	Grate area.	First 30 hours' trial. Indicated horse-power.	Speed.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.	Second 30 hours' trial. Indicated horse-power.	Speed.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.	Eight hours' full-power trial. Indicated horse-power.	Speed.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.
				tons		*sq. ft*	*sq. ft*		*knots*	*lb*	*lb.*		*knots*	*lb.*	*lb.*		*knots*	*lb.*	*lb.*
Dreadnought	Battleship	Portsmouth Dockyard	Vickers Sons and Maxim, Limited	17,900	Babcock & Wilcox	55,400	1,599	5,013*	13.00 log.	2.56	27.2†	16,930*	19.3 log.	1.7	17.01	24,712*	21 25 log.	1.51	15.56
Britannia.....	Ditto	Portsmouth Yard	Humphrys, Tennant and Co.	16,350	Babcock & Wilcox and cylindrical	48,291	1,483	3,539	11.44 "	2.05	21.0	13,087	16.79 "	1.5	16 21	18,725	18 4 "	1.83	18.55
Africa.........	Ditto	Chatham Yard	J. Brown and Co.	16,350	Ditto	48,396	1,490	3,682	11.8 "	2 001	...	12,860	17.5 "	1.793	...	18,698	19 1 "	1.872	...
Hibernia......	Ditto	Devonport Yard	Harland and Wolff	16,350	Ditto	48,648	1,476	3,710	10.4 "	2 07	19.9	12,700	15.5 "	1.59	17.41	18,112	18.127 M.M.	1.92	19 66
Natal..........	First-class cruiser	Vickers Sons and Maxim	Vickers Sons and Maxim, Limited	13,550	Yarrow and cylindrical	66,836	1,298	4,913	14.18 by bearings	2 03	20.1	15,937	21 109 M.M.	1.88	17.8	23.592	23.334 "	2.21	18.47
Cochrane....	Ditto	Fairfield Shipbuilding Company	Fairfield Engineering Company	13,550	Ditto	66,836	1,336	4,911	14.3 M.M.	2 06	19.1	16,080	21.37 "	2.04	16.4	23,654	23.292 "	1.99	16.4
Achilles......	Ditto	Armstrong, Whitworth, and Co.	Hawthorn, Leslie and Co.	13,550	Ditto	66,838	1,332	4,882	14 694 "	1.88	19 95	16,009	21.58 "	1.85	17.87	23,968	23 5 by bearings	2.03	19.9

* Shaft horsepower. † Consumption for ten hours. M.M. = Measured Mile.

H.M.S. "Gnat," the third to be completed of five coastal destroyers ordered from John I. Thornycroft & Co., Ltd., in connection with last year's Naval Program, was launched December 8, 1906, with all machinery on board, from the Company's Chiswick works. The dimensions of the class to which the *Gnat* belongs are: length, 168 feet; beam, 17 feet 6 inches; draught, 5 feet 11 inches; and the contract speed is 26 knots. She is fitted with turbine machinery of Parsons type, built at the firm's Southampton works, and Thornycroft water-tube boilers. The armament will consist of two 12-pounder quick-firing guns, three torpedo tubes.

New English Torpedo Launch "Dragonfly."—A forty-foot torpedo launch recently built in England is a most interesting craft, having been designed to operate at a speed of 18 knots. The *Dragonfly* is constructed to carry a 14-inch Whitehead torpedo, which is to be lowered over the side by means of a specially designed side drop gear. The body is first directed bow-on in the direction of the object which it is desired shall be destroyed, and then launched.

This torpedo boat is propelled by a 120-horsepower, gasoline engine, and is of 8,500 pounds displacement, with a draught of 2 feet 7 inches and a beam of 6 feet 2 inches. The boat is built of galvanized mild steel, and is provided with a watertight collision bulkhead. It has a very broad stern, which is said to overcome any difficulty as to stability when launching the torpedo over the side.

A turtle-back deck is arranged in the bow, extending to the rear as far as the after end of the engine, this being a portable piece, carried over the engine to protect it from the weather. Detachable spray guards are so placed as to prevent the splash from the bow waves being carried aboard.

The steering wheel and reversing lever are located on the port side. The engine is supplied with fuel from a tank which holds 100 gallons of gasoline, enough fuel, it is claimed, to operate the launch for a period of ten hours.

The *Dragonfly* is equipped with a four-cylinder engine which operates at a speed of 900 revolutions per minute. The

cylinders are of 8-inch bore and a stroke of 8 inches. The engine weighs about 2,500 pounds, which is at the rate of 23.25 pounds per brake horsepower. It is maintained that there is very little vibration, even at the speed of 900 revolutions per minute, the engine being of very light construction and very carefully and exactly balanced as to its moving parts. This is most essential, in fact absolutely necessary, on account of the very light construction of the hull, in order to insure high speeds and effectiveness in service. The engine is started by means of compressed air contained in a reservoir placed in the stern of the launch. A small Brotherhood compressor is provided for supplying the necessary air pressure, driven by a Thornycroft single-cylinder oil engine of 6-horsepower capacity.

The *Dragonfly* was constructed at the shops of John I. Thornycroft & Company, Limited, Chiswick, near London, where recently very interesting marine gas-producer outfits were constructed and installed in the launches *Emil Capitaine* and *Duchess of Chiswick*.—"International Marine Engineering."

The Steam Trials of the Armored Cruiser "Warrior."—H.M. armored cruiser *Warrior* has just completed her official steam trials, with satisfactory results. The *Warrior* belongs to the *Duke of Edinburgh* class of armored cruisers, being the sixth and last to be completed. She was built at Pembroke, and work on her was delayed owing to the withdrawal of the men required in connection with the salvage operations on the *Montagu*.

The ships of this class mark a great advance, not only in the protection aganst attack, but also in gunpowder; each cruiser mounts six 9.2-inch guns in separate barbettes—one on the forecastle, one on each side forward, one on each side toward the stern, and one in the center line on the upper deck aft. Four of these guns may be used on either broadside. There are also in the waist of the ship, on each broadside, two 7.5-inch guns, and the secondary armament includes twenty-six quick-firing guns for repelling torpedo attack.

The machinery of these ships was built to Admiralty design, and practically all the parts, including even the columns and bed plate, were made to standard gauges and jigs, in order to ensure, as far as possible, interchangeability. The main engines were designed to develop collectively 23,500 indicated horsepower, and the high-pressure cylinders were made 43½ inches in diameter, the intermediate cylinders 69 inches in diameter, and the two low-pressure cylinders, each 77 inches in diameter, the stroke in all cases being 42 inches. There are two large condensers to each engine, each having a cooling surface of 7,009 square feet, which serve for the main and auxiliary engines; either may do the work for each set of engines while the other is being overhauled. Some of the vessels have Babcock & Wilcox boilers, while others have Yarrow boilers, for four-fifths of the power to be developed, the remainder of the steam being supplied by cylindrical boilers. This combination, it will be remembered, was suggested by the Navy Boiler Committee, with a view of using cylindrical boilers for low powers. But as this combination has not shown any increased economy, and has involved a certain measure of inconvenience, the practice has been discontinued. The boiler installation in recent ships is entirely made up of water-tube boilers.

The machinery of the *Warrior* was constructed by the Wallsend Slipway and Engineering Company, Limited, and is the most important naval contract completed by them. It marks the progress of this company, which but a few years ago confined its operations to ordinary merchant work, but has, since the apppointment of Mr. Andrew Laing as managing director, stepped into the front rank, a fact indicated not only by the construction of the *Warrior's* engines, but by the order received for the 70,000 horsepower turbine machinery for the Cunard liner *Mauretania,* and still later by the contract for the turbine machinery for H.M. battleship *Superb.*

The first trial of the *Warrior* at one-fifth power was of 30 hours duration, and on this, as on all the trials, the vessel was loaded to the mean draught of 27 feet, while the boilers

maintained a pressure of 168 pounds to the square inch. The cut-off at the high-pressure cylinder was 42 per cent. of the stroke, and the engines, running at 83.2 revolutions, indicated 4,781 horsepower, the coal consumption working out at 2.01 pounds.

Table I.—STEAM TRIALS OF H. M. S. "WARRIOR."

Duration of trial	8 hours' full power.		30 hours' consumption trial at 15,700 I.H.P.		30 hours' consumption trial at 4,700 I.H.P.	
	Ford.	*Aft.*	*Ford.*	*Aft.*	*Ford.*	*Aft.*
Draught of water, feet and inches	26-00	27-06	26-07½	27-04¾	26-07½	27-04¾
Steam pressure in boilers, pounds per sq. in.	193.7		186.1		168.0	
Air pressure in stokeholds:						
Yarrow boilers, inches	1.1		0.4		...	
Cylindrical boilers, inch	1.0		0.4		...	
	Stbd.	*Port.*	*Stbd.*	*Port.*	*Stbd.*	*Port.*
Vacuum in condensers, inches	26.7	27.0	26.9	27.2	27.0	26 6
Revolutions per minute	137.7	137.5	121.4	121.2	83.0	83.4
Mean pressure in receivers:						
High, pounds	178.0	178.0	164.3	166.4	106.1	107.7
Intermediate, pounds	73.0	75.0	53.7	52.0	24.1	28.3
Low, pounds	24.0	23.0	10.5	11.3	1.0	1.5
Mean pressure in cylinders:						
High, pounds	65.9	72.6	68.5	71.37	30.47	30.5
Intermediate, pounds	35.7	32.7	27.7	25.11	10.61	12.14
Low, forward, pounds	18.0	18.5	12.26	12.05	5.3	5.12
Low, aft, pounds	19.1	19.2	12.51	12.17	5.21	4.71
Mean indicated horsepower:						
High	2,861	3,147	2,621	2,727	797	801
Intermediate	3,900	3,567	2,668	2,414	698	803
Low, forward	2,448	2,513	1,470	1,442	434	431
Low, aft	2,598	2,607	1,500	1,456	429	388
Total	11,807	11,834	8,259	8,039	2,358	2,423
Grand total	23,641		16,298		4,781	
Consumption of coal per indicated horsepower per hour, pounds	2.33		2.07		2 01	

The 30-hours' trial at two-thirds power was run in stormy weather, with a heavy sea, and these conditions militated strongly against economy. The air-pressure in the stokehold was 0.4 inch, and the steam pressure at the boilers 186

Table II.—MEAN RESULTS OF H. M. S. "WARRIOR'S" TRIALS.

......	At one-fifth power.	At seven-tenths power.	At full power.
Heating surface per indicated horsepower of main engines, square feet	7.38	4.1	2.83
Coal consumed per square foot of grate area, pounds	15.6	25.2	41.4
Indicated horsepower per square foot of grate area	7.8	12.2	17.8
Indicated horsepower per ton of machinery	...	...	10.8
Consumption of water per indicated horsepower per hour (for main engines and auxiliaries)	21.36	18.28	19.08

pounds. The high-pressure cylinder was slightly linked up, the average cut-off throughout the trial being 60 per cent. Running at 121.3 revolutions the power indicated was 16,298 horsepower, which gave the ship a speed of 20¾ knots, notwithstanding the heavy weather. The coal consumption worked out at 2.07 pounds, and, as shown in Table II, the water evaporated at 9 pounds per pound of coal.

The full-power trial was run on February 2 in favorable weather. The total power developed was 23,641 horsepower, when the engines were making 137.6 revolutions. This is equal to 10.8 indicated horsepower per ton of machinery. The air pressure in the boiler rooms was 1.05 inches. The coal consumed per square foot of grate surface was 41.4 pounds, while the indicated horsepower per square foot of grate was 17.8 horsepower. The average steam pressure at the boilers was 193.7 pounds. The results, which are set out in Tables I and II are therefore satisfactory, all the conditions of the contract being implemented.—"Engineering."

Oil Fuel for Marine Purposes.—Mr. Graydon Hume, in a paper read before the Institute of Marine Engineers, gave some useful practical information on the burning of liquid fuel at sea. He dealt with three methods, viz: first, using air as an atomizing agent; secondly, using steam; and, thirdly, atomizing by centrifugal action. The first involves the use of settling tanks, a pumping system, a meter for gauging the quantity of oil, air blowers, heaters, burners, and also a gasoline engine and auxiliary blowers. The main object of the tanks is to separate the oil from water held in suspension, and to this end there are wash plates in the bottom to prevent the water being again mixed with the oil when it has once settled. A steam coil is also fitted for heating the oil to from 100 degrees to 130 degrees Fahrenheit. The pumping system is for conveying the oil from the bunkers to the burners. Three pumps should be employed for alternate use, straining or filtering devices should have careful attention, and there should be an efficient air vessel. It is essential that all piping should be of iron, and not copper. With viscous oils there should be means of heat-

ing the oil in the bunkers; the usual manner being by a few turns of a steam pipe near the suction. Air blowers, generally of the Roots type, discharge into an equalizing pressure tank. The oil heaters are about 10 inches in diameter, and have a copper coil inside about 8 feet in diameter and 6 feet long; either live or exhaust steam can be used. The burners consist of a pipe through which the oil flows, and which is surrounded by a jacket which forms a passage for the air. There are two valves for respectively controlling the supply of air and oil.

The construction of the furnace does not, in Mr. Hume's opinion, get due attention. Complete combustion is impaired by too great a volume of air, and if the temperature of the gases is not sufficiently raised, smoke results, unless the quantity of oil is increased. In early stages of combustion the gases distil at low temperature and pass away unconsumed. Various arrangements of brick-work are used, but all agree that an extended front is necessary, and that the throat of the furnace should be lined. The bottom also should be of brickwork so as to cause any leakage of oil to be burnt. The back and sides should also be lined up to the lower row of tubes. Retarders in the tubes are beneficial. In an example from practice quoted the machinery of a ship developed 2,240 horsepower, and the consumption of oil was 1.08 pounds of oil per horsepower per hour, the funnel temperature being 540 degrees Fahrenheit. From a letter on the subject from a friend quoted by the author the following points may be summarized: Great care should be taken to have clean oil, as anything that will stop the regular flow will cause smoke, and that runs up consumption. The soot, being heavy, can not be blown away by steam, but must be swept; spring scrapers being best. Water can be detected by a "frying" sound. Heaters are essential with an air system, and make a steam system work better. With natural draft burners should point so as to converge about 3 feet from the burner end; and if draft is admitted below, they should be pointed down at a slight angle, otherwise the entering air drives the flame up against the crown of the furnace. A better way is to allow air to come in from the four corners of the

furnace, as this allows a minimum supply without smoking. The atomizing agent should not be used as a jet to increase the draft. Tubes should be swept every two months at least. Ferrules will stop tubes leaking, and have to be used when boilers are forced; they, however, retard the draft, and are, with small tubes, soon choked by oil soot. The oil pressure on burners should be kept as low as possible—about 15 pounds to 25 pounds—according to the density of the oil. With low pressure the valves can be run almost wide open.

Turning to the use of steam as an atomizing agent, Mr. Hume points out that large evaporators have to be installed to make good the water used. This will vary from 5 to 15 per cent. of the total evaporation. Examples are given of practice at sea with oil fuel with various descriptions of burners, the consumption ranging between 1.26 pounds and 1.48 pounds of oil per indicated horsepower per hour. The mechanical-spray system is described as similar in principle to the form of nozzles used for lawn sprinkling. Pressure is applied to make the oil flow and break it up. The passage of the fluid gives the required rotary motion, spiral blades being fitted for the purpose.

Upon a trial made with a Körting burner of this type, 14.9 pounds of water were evaporated per pound of Texan oil, from and at 212 degrees Fahrenheit.—"Engineering."

Return of Casualties to British War Ships shows that during 1905 eleven battle ships were damaged by accidents, the time of repair reaching a total of 580 days, or an average of 53 days apiece. Four armored cruisers were damaged by accident, and the repairs occupied 171 days, or an average of 43 days apiece; eight protected cruisers were damaged, and the repairs required 271 days, so that the average works out to 34 days. One cruiser was damaged and not repaired. Thirty-one destroyers, submarines and torpedo boats were damaged, the time of repairs averaging 43 days. Fifty-eight deaths and 39 cases of injury are recorded as the result of a total of 90 casualties to vessels. Thirty-three lives were lost on the occasion of the battle ship *Caesar* colliding with the barque *Afghanistan.* —"Army and Navy Register."

The Condition of Funnels of the British Ship "Argyll," as the result of overheating when steaming continuously at high speed during the recent maneuvers, has drawn attention to a weakness which has developed in the construction of the funnels of the latest British war vessels, due mainly to a desire to reduce the weights carried. The introduction of the water-tube boiler necessitated an increase in the number of funnels to ensure an efficient draft; at the same time, the heat given off, especially at high speeds, has greatly increased in intensity. Gradually, the number of funnels has been increased to four, while to keep down the weight the thickness of the casing has been fixed at three-thirty-seconds of an inch, and the funnel proper at three-sixteenths of an inch, and herein lies the weakness to which attention is drawn. The comparatively thin funnel plates, when exposed to the fierce heat generated at high speeds, generally erode, and from the experience gained in the *Argyll* and other vessels it is not too much to add that, if exposed continuously, as they would be in actual war, they would inevitably become not only a source of weakness, but of danger also to the ships carrying them. The remedy suggested is to increase the thickness of the plates, at least at the most exposed positions, to one-fourth inch, and we understand this is now receiving the attention of the Admiralty.—"Naval and Military Record."

EGYPT.

"Mahroussa."—An interesting piece of work has recently been completed by Messrs. A. & J. Inglis, Pointhouse, Glasgow, in the reconstruction of the Khedivial yacht *Mahroussa,* the work including the fitting of new boilers and turbine machinery in place of the original oscillating engines, the removal of the side paddle-wheels, and the reconstruction of the stern to suit triple-screw propulsion. The result has been most satisfactory, and the Khedive has himself paid a well-deserved compliment to the firm of Messrs. A. & J. Inglis. The *Mahroussa* was originally built forty-one years ago by Messrs. Samuda Brothers, for Ismail Pacha, and was probably at that date the

largest and most magnificent vessel of the kind in existence. Built with the best materials procurable, and with workmanship of the highest class, she was found on inspection after forty years' service to be much too good for the scrap heap, and sufficiently sound to be worth a complete refit. Sea-going paddle steamers, and low-pressure boilers with simple engines, are now, however, completely out of date, and the preservation of Penn's oscillating engines was not to be thought of. His Highness the Khedive of Egypt, being thoroughly well informed of what is going on in Western Europe, at first determined to have the *Mahroussa* transformed into a twin-screw steamer, and later, realizing the progress of the turbine, resolved to avail himself of the new motor. Messrs. A. & J. Inglis, Limited, of Pointhouse, Glasgow, had built for His Highness the 700-ton yacht *Safa-el-bahr,* which gave him the greatest satisfaction, and earned for Dr. Inglis the decoration of Commander of the Imperial Order of the Osmanieh, and as a consequence of the success of this yacht, they were asked to submit plans and estimates for the work to be carried out on the *Mahroussa* to Sir Vincent Corbett, K. C. V. O., the Financial Adviser to the Khedive, who called to his assistance Professor Biles, of Glasgow University, and, after some negotiations, the plans on which the alterations have been carried out were arranged.

The chief constructional work, so far as the hull was concerned, was asssociated with the adaptation of the stern to the triple-screw arrangement. This had to be carried out in a graving dock, and practically the only docks available for Messrs. Inglis were those of the Clyde Navigation Trust at Salterscroft. The trustees occasionally show some hesitation in granting the use of these docks for any lengthened period, as they are constantly in demand for cleaning and painting operations and urgent repairs to vessels frequenting the ports. It happened that an Allan liner, which had sustained serious damage, required a dock concurrently with the *Mahroussa,* and Messrs. Inglis were therefore urged to make their occupancy of the dock as brief as possible. To meet the Clyde Trust con-

dition, Messrs. Inglis devised special methods, so that the yacht would occupy the dock for the minimum of time. Molds were made of the frames while the vessel was afloat, and as much of the after-body as had to be reconstructed was laid down full size in the mold loft. From this draft the new stern frame, keel piece, balanced rudder, bossed frames, etc., were completed, ready to be erected as soon as the vessel was docked and the old stern cut away to receive the new. Every single piece of the new construction was found to fit so accrurately that when the job was completed there was nothing to show that the vessel had not been originally built as a triple-screw steamer. The engines and boilers had meanwhile been removed, the paddle boxes taken down, and the poop and deck-houses, as well as most of the decks, removed, bulk-heads shifted, bunkers rearranged, and new engine and boiler seats fitted. Steel decks were laid for about half the length of the ship. A range of deck houses, framed in steel and lined with polished teak, was erected on the upper deck, which was relaid with teak.

The new machinery, including the turbines, was constructed by Messrs. A. & J. Inglis, of Glasgow. It is the first work of the kind undertaken by the firm, and the success attained on the trial, on the voyage to Alexandria and on subsequent steaming, shows that this, one of the oldest of our engineering firms, still maintains its place amongst the most progressive.

There were in the old ship eight boilers of the square box type, the length being 15 feet 7 inches, and the width 10 feet 1 inch. These had each four furnaces, making thirty-two in all, the length of the firebox being 7 feet 6 inches, and the width 3 feet 4 inches. In all there were 3,696 tubes, of an external diameter of 3 inches, and having a length of 6 feet 6 inches, so that the total heating surface was 18,860 square feet. The boilers were designed to work at a pressure of 15 pounds. The new boilers, on the other hand, are five in number, of the cylindrical type, the length being 11 feet 4¼ inches and the diameter 14 feet 3 inches. In each boiler there are three furnaces, having an inside diameter of 3 feet 4 inches, with a length of fire bar of 7 feet 6 inches, giving a grate area of 300

square feet. The total number of tubes is 1,400, and they are 3 inches in external diameter and 7 feet 3¾ inches long, so that the total heating surface is 10,170 square feet. The working steam pressure of these boilers is 160 pounds.

As is now usual in moderate power installations, there are three turbines, the high-pressure turbine being mounted on the center shaft, with a low-pressure turbine on each wing shaft; each shaft has one propeller. The astern turbines are incorporated in the same casings as the low-pressure ahead turbines. The outside diameter of the high-pressure turbine casing is 5 feet 6 inches, and of the low-pressure casing 6 feet 11 inches. The blades are fixed on the principle adopted by Messrs. Parsons.

The astern turbines, it may be mentioned, are of unusual length, which should greatly increase the stopping and maneuvering power of the ship.

The speed desired with four-fifths of the boiler power was 16 knots, and this was exceeded on a trial of six hours' duration on the Clyde. When working at full power the mean of four runs over the measured mile at Skelmorlie was 17¾ knots, the engines running at 456 revolutions, and developing 5,000 shaft horsepower.

The yacht is now 400 feet long, 42 feet beam, and of 28 feet 2 inches molded depth; on a draught of 17 feet 2 inches the displacement is 4,300 tons. The vessel now has a very much smarter appearance than before. Originally, she had two funnels, and her immense paddle-wheels detracted from the grace of outline; the new vessel has only one funnel, and looks, as she practically is, a new vessel equipped on modern lines.—"Engineering."

GERMANY.

"Schleswig-Holstein."—This new German battleship was launched on December 17 at Krupp's Germania shipyard, Kiel, Germany. The following are the principal dimensions: Length between perpendiculars, 121.5 millimeters (398 feet 7 inches); greatest breadth, 22.2 millimeters (72 feet 10 inches); draught, 7.62 millimeters (25 feet).

Her belt armor has a thickness of 240 millimeters ($9\frac{7}{16}$ inches) amidships, tapering down to 100 millimeters ($3\frac{15}{16}$ inches) at both ends. The armored deck extends from stem to stern, and is curved down at the sides to the lower edge of the belt armor. The ship is fitted with an armored citadel and an armored gun-deck casemate. There are also two armored conning towers, the plates of which are 300 and 140 millimeters ($11\frac{13}{16}$ and $5\frac{1}{2}$ inches) thick. The armament consists of four 28-centimeter (11-inch) guns protected by turret armor 280 millimeters (11 inches) thick; ten 17-centimeter ($6\frac{11}{16}$-inch) guns protected by the casemate armor; four 17-centimeter ($6\frac{11}{16}$-inch) guns in separate casements; twenty 8.8-centimeter ($3\frac{7}{16}$-inch) guns; four 3.7-centimeter ($1\frac{7}{16}$-inch) quick-firing guns in the fighting tops; four 8-millimeter ($\frac{5}{16}$-inch) machine guns; six under-water torpedo-launching tubes.

There are three triple-expansion main engines of 16,000 indicated horsepower, supplied with steam by twelve multitubular Schulz boilers, and designed to give the ship a speed of 18 knots. The normal coal capacity is 700 tons; this can be increased to 1,600 tons by filling the reserve coal bunkers. The double-bottom compartments will contain 200 tons of coal-tar oil, which can be used for firing the boilers.

The tonnage of the *Schleswig-Holstein* is 13,200, and four other companion ships in the same class—the *Deutschland,* the *Hanover,* the *Pommern* and the *Schlesien*—are also under construction in German yards.—"Nautical Gazette."

ITALY.

Destroyers and Torpedo Boats for the Italian Navy.—In 1898 Messrs. C. and T. T. Pattison, of Naples, were commissioned by the Italian Government to build a number of 30-knot destroyers.

The design and working drawings of the hulls and machinery were supplied by Messrs. John I. Thornycroft & Co., Ltd., of Chiswick and Southampton, the boats being of a sim-

ilar type to those built by them for the British and Japanese navies.

The dimensions of this class are: Length 210 feet, beam 19 feet 6 inches, draught 7 feet 6 inches.

The engines are of the four-cylinder, triple-expansion, Thornycroft inclined, condensing type, each set having four cylinders as follows: High-pressure cylinder, 22 inches in diameter, intermediate cylinders, 29 inches in diameter, and two low-pressure cylinders, each 30 inches diameter. The stroke is 18 inches. The contract is for 6,000 I.H.P., and the designed boiler pressure is 220 pounds.

The armament consists of one 12-pounder quick-firing gun, and five 6-pounder quick-firing guns, and there are two torpedo tubes.

All the eight destroyers of this class have been launched and have fulfilled most satisfactorily the guaranteed conditions, each boat attaining a speed of at least thirty knots.

Messrs. Pattison also obtained orders for four first-class torpedo boats from designs supplied by Messrs. Thornycroft, which resembled boats recently built at Chiswick for the British Government except that twin screws were fitted instead of single screws. These boats also have gone through their trials with great success.

The mean speeds obtained on the three-hours' official trial of six of the torpedo boats were: *Pegaso,* 25.46 knots; *Perseo,* 25.72 knots; *Procione,* 26.02 knots; *Pallade,* 26.63 knots; *Cigno,* 26.40 knots; *Cassiopea,* 25.40 knots. The contract speed was 25 knots.

Subsequently to the placing of these orders the Italian Government called for tenders from Italian builders for destroyers and torpedo boats which were to be of Thornycroft type. Messrs. Ansaldo Armstrong & Co., of Genoa, and Messrs. Odero fu Alesso, of Sestri Ponente, became licensees of Messrs. Thornycroft, in Italy, and orders were placed with the former firm for four destroyers and with the latter for six torpedo boats, Messrs. Pattison also getting six additional torpedo boats.

In the design of the new destroyers Messrs. John I. Thornycroft & Co., Ltd., at the desire of the Italian Government, arranged for complete coal protection of the engines as well as the boilers of both types of boat. In the destroyers the engines are echeloned, and each engine is in a separate watertight compartment. The leading dimensions are: Length, 211 feet 6 inches; beam, 20 feet; draught, 7 feet 6 inches.

The engines are of the same type as those of the *Nembo* class already described, but the cylinders are 21 inches, 28½ inches, and two of 30 inches diameter by 18 inches stroke. There are, as in the *Nembo* class, three Thornycroft water-tube boilers. The armament consists of four 12-pounder quick-firing guns, and there are three torpedo tubes. The full complement of men is 53 in both the *Nembo* and *Bersagliere* classes.

It may here be mentioned that Messrs. Ansaldo Armstrong & Co. are also now building six stock destroyers identical in all respects with those of the *Bersagliere* class for the Italian Admiralty.

Twenty-four of the thirty Thornycroft water-tube boilers required for these ten destroyers have been or are being built by Messrs. Thornycroft at Chiswick, the remaining six being built, under license, by Messrs. Pattison at Naples.

The leading dimensions of the torpedo boats are as follows: Length, 164 feet; beam, 17 feet 4½ inches; draught, 7 feet.

The engines are of Thornycroft type and consist of two sets of vertical triple-expansion condensing engines with three cylinders 17 inches, 23½, and 36 inches in diameter, the stroke being 14 inches. They are capable of developing about 3,000 I.H.P.

There are two Thornycroft water-tube boilers in separate watertight compartments, which supply steam at 200 pounds pressure.

The armament consists of three 3-pounders and there are three torpedo tubes.

Three of these vessels have been tried and have exceeded their contract speed—the *Calliope* having attained a speed of 26.38 knots.

The destroyers and torpedo boats of Thornycroft design constructed for the Italian Navy may be seen at a glance from the following list:

Name.	Class.	Builders.	Launched.	I.H.P.	Speed.
Nembo....................	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Turbine....................	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Aquilone....................	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Borea	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Meteore	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Tuono	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Zeffiro....................	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Espero....................	Destroyer,	Pattison, Naples,	1901–2,	6,000,	30 knots.
Bersagliere....................	Destroyer,	Ansaldo, Genoa,	Building,	6,000,	28½ knots.
Artigliere....................	Destroyer,	Ansaldo, Genoa,	Building,	6,000,	28½ knots.
Lanciere....................	Destroyer,	Ansaldo, Genoa,	Building,	6,000,	28½ knots.
Granatiere	Destroyer,	Ansaldo, Genoa,	Building,	6,000,	28½ knots.
Alcione....................	Torpedo boat,	Odero, Sestri,	Building,	3,000,	25 knots.
Ardea....................	Torpedo boat,	Odero, Sestri,	Building,	3,000,	25 knots.
Albatross....................	Torpedo boat,	Odero, Sestri,	Building,	3,000,	25 knots.
Airone.	Torpedo boat,	Odero, Sestri,	Building,	3,000,	25 knots.
Astore....................	Torpedo boat,	Odero, Sestri,	Building,	3,000,	25 knots.
Arpia....................	Torpedo boat,	Odero, Sestri,	Building,	30,00,	25 knots.
Pegaso	Torpedo boat,	Pattison, Naples,	1905,	3,000,	25 knots.
Perseo....................	Torpedo boat,	Pattison, Naples,	1905,	3,000,	25 knots.
Procione	Torpedo boat,	Pattison, Naples,	1905,	3,000,	25 knots.
Pallade	Torpedo boat,	Pattison, Naples,	1906,	3,000,	25 knots.
Cigno....................	Torpedo boat,	Pattison, Naples,	1906,	3,000,	25 knots.
Cassiopea....................	Torpedo boat,	Pattison, Naples,	1906,	3,000,	25 knots.
Calliope....................	Torpedo boat,	Pattison, Naples,	Building,	3,000,	25 knots.
Clio....................	Torpedo boat,	Pattison, Naples,	Building,	3,000,	25 knots.
Centauro....................	Torpedo boat,	Pattison, Naples,	Building,	3,000,	25 knots.
Canopo.	Torpedo boat,	Pattison, Naples,	Building,	3,000,	25 knots.

—"Page's Weekly."

Submarines for Italy.—Three submarines have been completed for the Italian Navy—the *Delfino,* the *Glauco* and the *Squalo.* Three others are in course of construction—the *Narvalo,* the *Otario* and the *Tricheco.* The *Delfino* is rather an old craft, but she has experienced such a complete reconstruction that she may be practically considered a new boat. She has a displacement of 110 tons, and her length is 80 feet. She attained in her trials a speed of 8 knots. The *Glauco* and the *Squalo* are almost identical; their external aspect is that of a torpedo boat but they may be more properly called submersibles, while the *Delfino* is a submarine pure and simple. The *Glauco* and the *Squalo* have each a displacement of 150 tons; they are each 100 feet long. They attained in their trial a speed

of 13 knots. The *Glauco* plunged to a depth of 100 feet.—"Nautical Gazette."

Steam Trial of the Cruiser "Coronel Bolognesi."—The Peruvian cruiser *Coronel Bolognesi* completed at Barrow-in-Furness, on the 24th of January, her official steam trials, which were in all cases completely successful. The *Coronel Bolognesi*, with the *Almirante Grau*, have been constructed by Messrs. Vickers Sons & Maxim, Limited, for the Peruvian Navy. The *Coronel Bolognesi* differs only in having no half poop, which was adopted in the case of the *Almirante Grau*, as she is to be the flagship. The cruisers are 370 feet long, the beam being 40 feet 6 inches, with a draught of 14 feet 3 inches; the displacement is 3,200 tons. There is an armored deck throughout the whole length of the ship, which encloses the machinery compartments, magazines, steering gear, etc., while coal bunkers are arranged along the sides of the vessel to protect the vital parts of the mechanism. The conning tower has 3-inch armor. Each vessel carries a bow and a stern-chasing 6-inch gun, protected by a shield, and on each broadside there are four 14-pounder guns, which, with an installation of pom-poms, give a satisfactory offensive power for the displacement. The trials consisted of a full series of progressive-speed runs, in addition to full-power and progressive-power coal-consumption trials. During the full-speed trial the vessel made six runs over the measured mile, and attained a speed of 24.726 knots, the guaranteed rate being 24 knots. The engines, which are of the four-cylinder triple-expansion type, averaged 218.7 revolutions, the power developed being 14,384 indicated horse-power. Everything worked satisfactorily, and on the conclusion of this trial the required maneuvering tests were carried out, the vessel making circles at 14 and 18 knots. The vessel made a 24-hours' endurance test, which was divided into four successive periods of equal length, and the machinery was run at progressive powers. The results are tabulated, and alongside we give the performance at the corresponding trial of the *Almirante Grau*.

	Coronel Bolognesi.		*Almirante Grau.*	
	Indicated horsepower.	Coal consumption. Pounds per I.H.P.	Indicated horsepower.	Coal consumption. Pounds per I.H.P.
First six hours......	1,078	2.31	913	2.27
Second six hours...	3,058	1.66	3,689	1.45
Third six hours.....	6,764	1.71	6,342	1.68
Fourth six hours...	9,485	1.7	8,817	1.63

Admiral Carvajel, the head of the Naval Commission, was present at the trials, and determined the powers at which the successive coal-consumption trials were to be run; this accounts for the variation between the *Almirante Grau* and the *Coronel Bolognesi.* The Admiral is to be congratulated on the success of his work in connection with the construction of these two vessels, as they possess a high speed and good offensive and defensive powers for their size, with a large radius of action, the distance steamed at cruising speed per ton of coal burnt being between 8 and 9 miles. It is hoped that they will form a nucleus for a new navy for Peru; they are bound to awaken interest in naval affairs among the Peruvians, may prove a satisfactory training-school, and at the same time will, when called upon, give a good account of themselves in the protection of the Peruvian coast against heavier and more costly ships not endowed with the same speed.—"Engineering."

RUSSIA.

The Scheme for the Reconstruction of the Russian Navy is to build twelve battleships, fifteen cruisers, forty-six torpedo-boat destroyers, eighteen torpedo boats, ten submarines, seven gunboats, nine monitors, or coast-defense turret ships, and one mining ship. The building of these 118 vessels is to be spread over a term of nine years. The scheme has received the sanction of the Emperor, and as its details have been communicated to the Austrian press, the following account may be relied upon as correctly summarizing them:

The scheme provides for three fleets—the Baltic, the Black

Sea and the Far Eastern; whilst one stationary gunboat is told off to the Persian Gulf. For the Baltic fleet, headquarters Cronstadt, there will be built nine turret battleships, each of from 16,000 to 17,000 tons displacement, with 4 heavy (presumably 12-inch), 14 medium (probably 7¾ to 9-inch), and 56 quick-firing guns each. These ships are to be armored with nickel-steel plates 9 inches thick, and to have a speed of 18.5 knots. It will strike experts that for battleships of the future these vessels are defective, in that they will be inferior in heavy guns to any foreign ships that they might meet, except those already constructed and almost out of date. They will also be of less displacement. Four first-class cruisers of 12,000 displacement with a complete armor belt, and 5¼ inches protection to the central battery, will not carry any 12-inch guns, but are designed to accommodate 40 quick-firing guns each. There will be eighteen deep-sea torpedo boats of 250 to 300 tons each, a mining ship like the *Yenesei,* blown up near Port Arthur, and ten submarines. This concludes the Baltic fleet program.

In the Black Sea fleet the obsolete turret ships *Sinope, Ekaterina II* and *Tchesma* are to be replaced by three armored turret ships of 12,500 tons displacement, each with 4 12-inch guns, 4 7¾-inch and 12 6-inch, besides 32 small-bore quick-firing guns. These ships will carry 9-inch armor and will steam 16 knots an hour. This style of ship is not good enough for twentieth century requirements, though powerful in its somewhat antiquated way. Seven first-class cruisers are to be built for the Black Sea fleet, each of 12,750 tons, armament not yet specified; four second-class cruisers of the *Otchakoff* type, of 6,700 tons displacement, and a speed of 22 to 24 knots. These should be very useful craft. Twenty-eight torpedo boats of the *Svonki* (350 tons, 26 knots speed) type complete the list for the Black Sea. This fleet will, therefore, be in the next, as in the last, decennial period altogether behind the times as to material, which is constructed to meet past or, at most, present requirements only.

For the Far East are to be built six coast gunboats of 800 to 1,000 tons each, armed with 1 heavy, 2 medium, and 12

light guns, and a 14-knot speed; also nine shallow-draught gunboats of 250 tons each for the Amur and Sungari rivers. The total cost of this building scheme is to be 380,000,000 roubles or £38,000,000, divided into nine annual instalments, which will be added to the normal annual expenditure of 117,-000,000 roubles (£11,700,000). Thus, the total annual naval budget for the next nine years will be 159,200,000 roubles (£15,920,000), which will include cost of equipment, arsenal and dock construction, etc., etc. Considering the great losses suffered by the Russian Navy during the war with Japan, this cannot be considered an excessive amount to be spent on re-forming it, which is practically what has to be done. It is evident that the Russian Admiralty is not prepared as yet to lay down the ships which, according to our ideas, are now nec-essary—that is, battleships of nearly 20,000 tons displacement, and an armament of more than 4 12-inch guns. Of course, the future will decide who is right; we are, ourselves, on the side of the heavily-armed ship of considerable speed and coal radius, able to meet anything floating on the ocean.—"The Broad Arrow."

JAPAN.

Launch of the "Satsuma."—One year and one month after the peace of Portsmouth, which was brought about by the noble efforts of President Roosevelt, the launch of the largest battleship afloat took place in the presence of H. M. the Emperor, the Crown Prince, many princes and princesses, and a huge number of all classes of people, at the Yokosuka navy yard, which is but five miles from Uraga, where the monument to Commodore Perry stands.

The battleship *Satsuma,* the construction of which began in the midst of the Russo-Japanese war, is 482 feet in length, 83 feet 6 inches in beam, of 19,200 tons displacement and 18,000 horsepower. Her armament is not yet officially declared, and will be kept secret until completion. But the authorities, it is said, at first intended to provide four 12-inch guns, twelve 10-inch guns, twelve 4.7-inch guns, and five torpedo tubes. Thus it will be seen that Japan has not dispensed with intermediate

armament, as is the case with the *Dreadnought.* Incessant progress in naval matters, however, calls for some new alterations and improvements to be introduced to the armament; and the *Satsuma* will, it is believed, be finally found to be more powerfully equipped than was originally intended. Her armor belt of Krupp steel ranges from 5 to 9 (or 9½) inches, and her intended speed is 19 knots. The ram bow has been dispensed with in her, as in the two armored cruisers, *Tsukuba* and *Ikoma,* just built respectively at Kure and Yokosuka. She has a very handsome semi-fiddle bow. Over a year ago, Admiral Sir Cyprian Bridge said it would be interesting to see how long the ram bow would be a feature of warship design. So far as the Japanese are concerned, the day of the ram has passed away, and will not be revived in our future warships unless some development, as yet undiscovered, is made hereafter in naval warfare. When the *Satsuma* is fully equipped she will also be without the fighting tops so common in modern warships. Compared with our latest battleship, *Kashima*, she has a larger displacement by 2,600 tons, and in armament has eight more 10-inch guns. Not only is the *Satsuma* much superior to the *Kashima* in her exterior design, but the difference in her interior design is incomparably greater, owing to the fact that in the construction of the *Satsuma* every available experience obtained from the late war has been turned to account. The new battleship has a larger displacement than the *Dreadnought* by 1,300 tons, though she is inferior in point of speed; and there is a question as to the comparative strength of the two battleships' armaments. The *Satsuma* has four 12-inch and twelve 10-inch guns against the *Dreadnought's* ten 12-inch, so that in fire the latter opposes six 12-inch to the former ship's twelve 10-inch. The allied nations are to be congratulated upon their possession of the two most powerful battleships in the world. In the construction of warships, the most valuable of all experiences are undoubtedly those derived from the tests of actual engagements. A battleship, designed by the experts of a country which has had various experiences of modern naval warfare,

cannot fail to have many characteristics peculiar to itself; though the public are yet in the dark as to the details of those characteristics.

On November 15, when the launch had been arranged to take place, His Majesty entered the imperial stand at about 2 P. M., which faced the stem of the ship. Preparations for the launch were soon commenced. The shores supporting both sides of the hull, the wedges, etc., were removed in accordance with signal orders Nos. 1 to 14. The Minister of the Navy, Vice-Admiral Saito, then proceeded before the throne and read the following document: "On the 15th day of May in the 38th year of Meiji (1905) the construction of the battleship numbered B was commenced, and the hull having now been completed, His Majesty is pleased to name her *Satsuma*. The Minister handed the document to Vice-Admiral Kamimura, commander of the Yokosuka naval station, and the latter immediately instructed the superintendent of the arsenal, Vice-Admiral Ito, to launch the ship. As soon as the cord was cut by Vice-Admiral Ito, the hull began sliding. As the *Satsuma* was smoothly going down toward the water, a ball hanging from her bow was automatically broken, scattering pieces of colored paper, cloth, flowers, etc., from among which several pigeons flew away. The thunderous *Banzai* and applause continued for a time. The ship was entirely afloat at 2:25 P. M. It may be added that the *Satsuma* has been built entirely by Japanese experts, and there is no truth whatever in the reports circulated in Europe as to a number of foreign engineers having been employed. SAITO TSUNETARO,

The Imperial Fisheries Institute, Etchujima, Tokio.

—"Scientific American."

OBITUARY.

BENJAMIN HOWARD WARREN.

By a Member.

It is probable that the general public would hardly count the Naval Academy as a training school for business men, although an examination of the Graduates' Register would show a not insignificant percentage of Annapolis men who have made a fine record in business life, and among these Mr. Warren occupied a notable place. His active naval career was short, lasting only from 1871 to 1878, but he was one of the most loyal sons of Annapolis and his interest in naval matters was very sincere all through his life.

He was a member of the first class of Cadet-Engineers, which entered in 1871, but did not graduate until 1874 on account of enforced absence due to illness. After a short sea service, he was retired in 1878 on account of partial deafness. This covers his active naval life except for a short time in 1898, during the Spanish War, when he was chief steel inspector in the Pittsburg district, the Government getting his services for less than a tenth of his salary in civil life.

Mr. Warren's whole life was spent as an engineer or an executive in engineering works. At first he was with the Hancock Inspirator Co. in various capacities, as mechanical engineer, European representative, and general manager. Then for some six years he was manager of the crane department of the Yale & Towne Mfg. Co., severing his connection when that department was sold to the Brown Hoisting Co., of Cleveland. For a short time he was with the Pratt and Whitney Co., and then joined the Westinghouse Electric Co.

It was during his service with this company that Mr. Warren made his great reputation as an executive. He began as assistant general manager in charge of manufacturing, and

displayed such remarkable talents as an organizer that Geo. Westinghouse, the head of the great company, made him a vice-president in charge of manufacturing and commercial operations. The growth of the company during his administration was phenomenal, and a large share of the credit for its success has always been given to Mr. Warren. He was a rigid disciplinarian, stern and sometimes severe, but scrupulously fair and just to everybody. At the end of 1901 he resigned his position and devoted himself to the care of a moderate fortune which he had accumulated, and to the enjoyment of a beautiful estate he had purchased near Charlottesville, Va. In 1904 he was invited to become the President of the Allis-Chalmers Co., another great manufacturing concern, a position which he filled until September, 1905, when he resigned on account of impaired health. In the early part of this year he became a partner in the firm of Kafer, Mattice & Warren, consulting engineers. His death on October 20th from apoplexy, in his 57th year, was totally unexpected.

Mr. Warren was a man of warm affections, although not demonstrative, and sincerely devoted to any cause in which he was interested. He was very active in the New England Association of Naval Engineers, frequently making the journey of seven hundred miles from Pittsburg to attend the annual dinner. For a time he was the owner of "The Engineer" of New York, and opened its columns freely to the discussion of naval matters. He was also the publisher of Bennett's History of the Steam Navy of the United States. By inheritance from his father, who was an officer of the Army in the Civil War (killed at the battle of the Wilderness), he was a companion of the Loyal Legion. He had been a vice-president of the Society of Mechanical Engineers, and was also a member of the Society of Naval Architects and Marine Engineers. He was an early member of the Engineers Club of New York and always very active in its affairs. He leaves a widow, two daughters and a son, and his was the first death in a family circle of rare charm and one which it was a privilege to enter as a friend. He had a very wide acquaintance in

engineering and manufacturing circles and counted among his personal friends most of the leaders in the profession.

It was a remarkable fact that his death is the first among the engineer graduates of 1874.

CHARLES HARDING LORING.

BY W. M. MCFARLAND, MEMBER.

On February 5th there passed away after a long and distinguished life a man who had always upheld the highest traditions of the Service and who, under all circumstances, has been one of its ornaments, Charles Harding Loring, Chief Engineer, with the rank of Rear Admiral. All the highest honors of the engineering profession had been conferred upon him, both on the scientific and social sides, thus bearing marked tribute to his eminence as an engineer and to his sterling worth and personal charm as a man.

He was born in Boston December 26, 1828, and received his education in the public schools of that city. As his educational period was before the day of technical schools, he followed the usual course of preparation for mechanical engineering and served a regular apprenticeship in the machine shop. At its close, in 1851, he entered the Navy as a third assistant engineer, attaining, by competitive examination, the highest place in a class of fourteen.

His entrance was just too late to give him an opportunity for participation in the Mexican War, and by the time the Civil War broke out he had passed through all the junior grades and had become a chief engineer. During his service in junior grades he had been laying the foundation for his more important work when an older man and in higher positions, a portion of his shore duty having been as assistant to the Engineer-in-Chief of the Navy, Mr. Samuel Archbold, in which capacity he had charge of the experimental work and tests of engineering devices coming before that office. It is interesting to note that while engaged in this duty he made a test of the first injector which came to this country.

During the Civil War he was in active service the whole time, and during the first eighteen months was fleet engineer of the North Atlantic Station, being attached to the fine old steamer *Minnesota*. He was on board this ship during the attacks of the *Merrimac* on the Northern fleet in Hampton Roads on the eighth and ninth of March, 1862, when the *Cumberland* was sunk and the *Congress* burned, and when the *Minnesota* also was attacked.

A little later he was detached from sea duty and sent to Cincinnati to supervise the construction of three river and harbor monitors and also of some light-draught sea monitors building there. Subsequently he was made general inspector of all the iron-clad steamers building west of the Alleghanies, having in charge at one time eleven monitors building at Pittsburg, Cincinnati and St. Louis.

During the Civil War a number of excellent engines had been accumulated for hulls which were in process of construction, but with the close of the war all work was stopped, and after a time a board was appointed to recommend the best disposition of these engines which were stored in the various navy yards. It was about this time that the compound engine was coming into general use, and the same board was directed to make a study of the compound engine with a view to its introduction in naval vessels. Of this board Admiral Loring was senior member, and associated with him was the late Chief Engineer Charles H. Baker.

After a very exhaustive study of the subject, they recommended the introduction of compound engines and the abandonment of the simple form, and the conversion of a number of the engines which were on hand into compound engines. Four sets of these simple engines were so converted and were fitted to the *Vandalia, Marion, Quinnebaug* and *Swatara*. The tests of these engines were very satisfactory and showed a coal economy for short runs of not much over two pounds of coal per horsepower hour.

This study of the compound engine made it natural that Admiral Loring should be selected as the representative of the

Navy Department when, in 1874, he and the late Dr. Charles E. Emery made an elaborate series of trials of the engines of the revenue cutters *Rush, Dexter, Dallas* and *Gallatin,* to determine by actual test the relative economies of compound and simple engines, designed for the same work in similar hulls, and also to secure reliable and authoritative data with respect to the economy of steam jacketing. These tests were the first of the kind conducted under circumstances of entire reliability, with the result that the report of the trials was republished all over the world and is still quoted in all the text books on steam engineering.

Admiral Loring's next tour of sea duty was as a fleet engineer of the Asiatic Station on the U. S. S. *Tennessee,* where he had as his chief assistant, George W. Melville, who later became his successor as chief of the Bureau of Steam Engineering. There was nothing specially eventful in this cruise, and at its end, in 1880, he was assigned as the head of the steam engineering department of the New York Navy Yard.

This was the period of greatest inactivity in the history of the Navy, and there was little to do, even for a very active man, except routine work. During this tour, however, Admiral Loring was senior member of a board that made a test of the machinery of the *Anthracite,* a little yacht with a triple-expansion engine working with 600 pounds pressure. The experiments were valuable as showing that, with the form of apparatus on board the *Anthracite,* there was no such gain in economy as to warrant the tremendous pressure carried, while it involved numerous practical difficulties.

In 1881 he was a member of what is known as the "First Naval Advisory Board," appointed by Secretary Hunt to formulate a shipbuilding program for the Navy which he might submit to Congress. The personnel of this advisory board was distinguished in all its branches, and the work they did made possible our splendid fleet of today, as they definitely decided to abandon wooden hulls for those of iron and steel, and for general progress in every respect. In 1882 he was a member of another important board known as the "Navy Yard

Board," of which Admiral Luce was senior member. The duty of this board was to visit all the navy yards of the country for the purpose of determining which of them might with advantage and economy be closed. It was a delicate task, but the report, when finally approved, gave general satisfaction, and its recommendations were carried out.

On the retirement of Engineer-in-Chief Shock only two successors were thought of, one of whom was Admiral Loring, and his merit and thorough qualification for the position were so well recognized that the appointment came to him entirely unsought. This was in 1884, during the administration of President Arthur. Secretary Chandler was presiding over the Navy Department at this time, and it was under his supervision that the four vessels commonly known as the Roach cruisers, the *Atlanta, Boston, Chicago* and *Dolphin,* were built.

Part of the scheme of building the new navy was the organization of what was known as an "Advisory Board," composed of two civilians and a number of naval officers. Owing to this régimé the bureaus were not given the same free hand that has obtained since the advisory board was discontinued, although they did valuable work in the details of designs. Forced draft was used on these new vessels, after having been tried on two others—the *Alliance* and *Swatara*—under Admiral Loring's direction.

In 1885, with the advent of a new administration, there was a general spirit of unrest about the Navy Department from what seemed to be a prevailing belief that whatever was, was wrong. The air was filled with rumors of intended changes, among them one which promised to cause a violation of the contract labor law, as it was actually seriously under consideration to import a British engineer and put him in charge of the design of machinery. From this intent and other indications it became evident to Admiral Loring that he did not enjoy the Secretary's confidence, and he tendered his resignation.

After leaving the Bureau of Steam Engineering he was

made senior member of the Experimental Board of Naval Officers at the New York Navy Yard, which board, under his direction, conducted many exceedingly valuable experiments. Among the most important were the competitive tests of water-tube boilers to determine the type that should be used on the coast-defence vessel *Monterey*, and it may be well here to call attention to the fact that this was the first case on record where a boiler had ever been run for twenty-four hours when burning more than fifty pounds of coal per square foot of grate.

Another very important series of experiments conducted by Admiral Loring were those on the boilers of the torpedo boat *Cushing*, to determine the economy of evaporation with different air pressures and rates of combustion. These experiments have proven of the greatest interest, and form a very valuable collection of engineering data. A number of clever devices had to be schemed out to carry on these tests, and the whole success was a great credit to Admiral Loring and the board.

Having reached the age limit in December, 1890, he was placed on the retired list; but having always been a man of very vigorous physique, he did not give up active employment, and was for a time consulting engineer to the United States and Brazil Mail Steamship Company. During the late war with Spain he was recalled to active duty and assigned as inspector of engineering work in New York City.

This record shows the active naval work of Admiral Loring, but to those who knew him well, this side of his character was less important than the social one. Although a man of great dignity, he was what is now called "a good mixer," being most companionable and a delightful associate. He wrote well and was a good speaker, but he was perhaps at his best in a party of moderate size where his keen sense of humor, his genial personality and his remarkable skill as a raconteur made him a most enjoyable associate.

He was for two years president of the Engineers' Club of New York, which is a sure test of popularity and wide ac-

quaintance in the engineering world. He was also president of the American Society of Mechanical Engineers, the highest honor which his branch of the profession could confer. He was a vice-president of the Society of Naval Architects and Marine Engineers from its formation until his death, and, as long as his health permitted, was very active in its council and general meetings. He was a member of our own Society from the beginning. He had also been very active in the Army and Navy Club of New York, filling various offices and acting as its secretary for several years.

Time had dealt gently with his appearance, and although in his seventy-ninth year when he died, he did not look much over sixty. Until the last few years he had been very robust, but he had been ailing for some time and finally succumbed to an attack of paralysis.

Admiral Loring's career covered almost the whole period of progress in marine engineering from the side-wheel engine and box boilers down to the triple-expansion engine and steam turbine, forced draft and the water-tube boiler. In nearly all of this progress he played an important part, and he could justly feel that he had not only done his duty but had been a factor in the continued advancement of the profession and the service he loved so well.

ALFRED BRUCE CANAGA.

BY A MEMBER.

It is not often that the death of an officer causes such general and sincere sorrow as that of Commander Canaga, who was affectionately called by his intimate friends, "Pop." His character and disposition were so admirable and uniform that no one could be associated with him without becoming a warm friend. The well-known lines about Abou Ben Adhem tell us that he was given a high place in the good book because he loved his fellow men. Surely dear old "Pop" will rank equally high. His kindness and consideration for others and his help-

fulness to all his associates make his record one to be admired but almost impossible to equal.

Commander Canaga was an honor man in the second class of Cadet Engineers, graduating in 1874. It is rather remarkable to note that the ranks of the ten graduates of this class had not been broken until the death of Mr. Warren in October, followed so soon by that of Commander Canaga on December 24, 1906.

Of his active naval life, about half (some sixteen years) was spent at sea, covering almost every part of the globe. It happened, too, as a common thing with him for his cruises to extend beyond the usual three years. His shore duty was all of a very important nature and commensurate with his high order of ability. He was twice on duty at the Bureau of Steam Engineering as the chief designer, in charge of the drawing room, during which time many of the designs for our most important vessels of the new navy were worked out. He also did some important educational work, as an instructor at the Naval Academy and as a professor at Cornell University. He had been Chief Engineer of the Cavite Dock Yard, and was on duty as Chief Engineer of the Boston Navy Yard at the time of his death.

Such is a brief statement of a very busy life, full of work, always well done. A man of high character and unblemished reputation, he leaves a record which is a source of just pride to his family and of affectionate remembrance by his host of friends. He had just completed his 56th year, having been born November 2, 1850.

He leaves a widow, a daughter, and a son who is an Ensign in the Navy.

ASSOCIATION NOTES.

The regular annual meeting of the Society was held at the Navy Department, Washington, D. C., on December 27, 1906, for the purpose of electing officers for 1907.

A count of the votes received resulted in the election of the following officers:

President, Commander B. C. Bryan, U. S. Navy.

Secretary-Treasurer, Commander Theo. C. Fenton, U. S. Navy (retired).

Members of the Council, Captain A. F. Dixon, U. S. Navy; Commander R. S. Griffin, U. S. Navy; Commander H. P. Norton, U. S. Navy.

The following is the statement of the Secretary-Treasurer for the year ending Dec. 31, 1906:

Received, 1906:		
Dues and subscriptions	$3,911.35	
Advertisements	1,834.87	
Sales	544.73	
Interest	184.73	
Total receipts		6,475.68
Balance received Jan. 1, 1906		5,746.63
		$12,222.31
Expended:		
Printing and stationery	$4,319.29	
Engraving	685.05	
Salary Secretary-Treasurer	900.00	
Purchase of JOURNALS	12.50	
Articles paid for	100.00	
Postage, expressage, incidentals	72.00	
Purchase of five (5) $1,000 bonds, Washington Railway and Electric Co	4,400.00	
Total expended		$10,488.84
Cash balance		$1,733.47

Available assets of the Society, January 1, 1907, cash balance...	1,733.47
Five (5) $1,000 bonds (cost)........	4,400.00
Total........	$6,133.47
Total value, Jan. 1, 1906, cash........	5,746.63
Net gain for year 1906........	$386.84

This statement was audited by a committee of the Council and found to be correct.

THEO. C. FENTON, *Commander*, *U. S. N.*, *Retired*,
Secretary-Treasurer.

B. C. BRYAN, *Commander*, *U. S. N.*,
President.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XIX. MAY, 1907. No. 2.

SOME NOTES ON GAS ENGINES.

BY PROF. H. DIEDERICHS, CORNELL UNIVERSITY.

[From *The Sibley Journal of Engineering*.]

If there is one movement of extreme importance in the engineering field of today it is the increased interest manifested in the gas engine as a source of motive power. Opinions as to the probable extent of this movement are various. Some will grudgingly concede that the gas engine is quite well enough developed to compete with the reciprocating steam engine in some fields, others, among them Diesel, claim with great assurance that the death knell of the steam engine has been sounded, and that it is but a matter of a few years when the steam engine will be a thing of the past. The impartial observer can not well agree with either the one or the other of these views. It is unquestionably a fact that the internal-combustion engine can show an immense improvement in the last ten years, and not the least among these the steps in advance, is the fact that gas engines can now be built for successfully operating alternating-current machinery in parallel, but to conclude from

this showing as a whole that the steam engine is absolutely doomed would seem to be going too far. It can, of course, not be denied that the internal-combustion motor will find much wider application than it now has, but it is quite safe to conclude that the conditions and exigencies of every case will decide between the gas engine and the steam engine, much in the same way as we decide today between high and low-speed, single or multiple-cylinder, condensing or non-condensing engines.

Practically every important industrial nation is concerned in this advance. It appears that we in America have been somewhat behind hand, and have allowed German and English manufacturers and designers to show us the way in some branches of this development, but, unless the signs fail, this state of things will not continue for any considerable length of time.

As an evidence of this movement, and in a measure of its importance, we find a large amount of scientific and practical discussion appearing in scientific journals and papers. In many cases these articles are of great importance and wide bearing, and this is especially true when they come from the pens of such men as Humphrey and Burstall in England, Witz and Hubert in France and Belgium, and Meyer, Schöttler, Schröter, Güldner, Diesel and others in Germany. It is not an easy matter for any engineer to keep himself posted and abreast of the times on engineering matters concerning his own country. Much more difficult is the task when he attempts to do the same thing in a foreign language with which he may be little familiar. Nevertheless the importance of this subject of gas-engine development and the fact that some other nations at the time of this writing have admittedly a slight lead in this field, would justify some pains in this direction.

It is the purpose of the following paragraphs to report upon and discuss briefly some of the interesting tests and treatises that have appeared in German literature within the last two years, together with some data which have come under the writer's own observations. After what has been said above it is hardly necessary to further justify the presentation of this

matter in this shape, and while no doubt much of the following is familiar to some, it is hoped that some service may be done.

I. GAS ENGINE CYCLES, THEIR THEORETICAL THERMAL EFFICIENCIES AND PRACTICAL LIMITATIONS.

On examination it will be found that all practical internal-combustion motors work with one or the other of two cycles. The first class employs the Otto cycle, the second a cycle the type of which is the Diesel cycle of today. In our discussion we will call this the Diesel Cycle. This classification holds good whether the engines employ the 4-cycle or 2-cycle principle. Since the Diesel of today is a 4-cycle engine, we will further limit this discussion to the 4-cycle engine, it being understood, however, that the thermodynamic expressions derived hold just as well for the 2-cycle type.

I. THE OTTO CYCLE.

In this cycle heat is added along the line 1—2 at constant volume. Adiabatic expansion takes place along 2—3. The heat is rejected along line 3—4 at constant volume. The remainder of the charge is displaced along 4—5, while a new charge is drawn in along 5—4. This is adiabatically compressed along 4—1, completing the cycle. In the theoretical cycle, the work along 4—5 equals that along 5—4, and since neither of these changes have any connection with the heat interchanges in the cycle proper, the cycle to be examined consists of the lines 4–1–2–3–4.

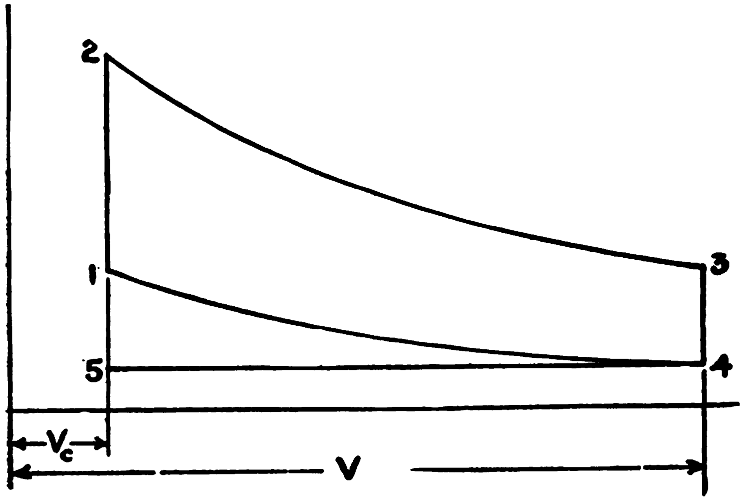
Let H_s be the heat supplied to the cycle.

Let H_r be the heat rejected.

Let $p_1, p_2, \cdots, v_1, v_2, \cdots T_1, T_2, \cdots$, be the absolute pressures, volumes and absolute temperatures, respectively, at the points 1, 2, $\cdots$.

Finally, if K_p and K_v be the specific heats of the charge at constant pressure and constant volume, respectively,

$$\text{let } x = \frac{K_p}{K_v}.$$



The cyclic, or the possible thermal efficiency of any cycle, is

$$E = \frac{H_s - H_r}{H_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

Substituting in (1), the values of H_s and H_r determined from well-known thermodynamic conditions for this cycle, we shall have

$$E = \frac{(T_2 - T_1) - (T_3 - T_4)}{T_2 - T_1}$$

$$= \frac{T_2\left(1 - \frac{T_3}{T_2}\right) - T_1\left(1 - \frac{T_4}{T_1}\right)}{T_2 - T_1}.$$

$$\text{But } \frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ from which } \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

$$\therefore E = \frac{(T_2 - T_1)\left(1 - \frac{T_4}{T_1}\right)}{T_2 - T_1} = 1 - \frac{T_4}{T_1} = 1 - \frac{T_3}{T_2}.$$

This may also be written

$$E = 1 - \left(\frac{V_c}{V}\right)^{x-1}$$

If we let $\dfrac{V}{V_c} = \dfrac{\text{total volume at end of stroke}}{\text{clearance volume}} = r$ we shall finally have

$$E = 1 - \left(\frac{1}{r}\right)^{x-1} = 1 - \frac{1}{r^{x-1}}.$$

This expression shows that the efficiency of this cycle depends upon the values of r and x. It shows that as r is increased, *i. e.*, as the clearance volume is decreased, the efficiency will increase. Further, as x increases the efficiency will also increase. Now x depends upon the composition of the mixture, for any given fuel, gas or oil, x being lower the richer the charge. So that this expression finally shows that the leaner fuel mixtures may be expected to give the better results.

This fact is well illustrated in the following figures.*

Mixture of *Ill. Gas*	*Air*	$r=$	$x=$	$E=$
1	6	5	1.354	.435
1	13.5	5	1.383	.461

This shows a difference of 6 per cent. in favor of the poorer gas mixture.

Lastly, since the efficiency depends only upon r and x, it is independent of the maximum temperature in the cycle, *i. e.*, it is independent of the load on the engine as long as r and x remain the same.

II. THE DIESEL CYCLE OF TODAY.

In this cycle we have heat added along 1—2 at constant pressure, and adiabatic expansion from 2—3. The heat is rejected at constant volume along 3—4. The charge is displaced along 4—5, and a new charge drawn in along 5—4. The final step is adiabatic compression along 4—1.

The best treatment of this case would seem to be the following: †

As before
$$E = \frac{H_s - H_r}{H_s},$$
$$H_s = K_p (T_2 = T_1),$$
$$H_r = K_v (T_3 = T_4).$$

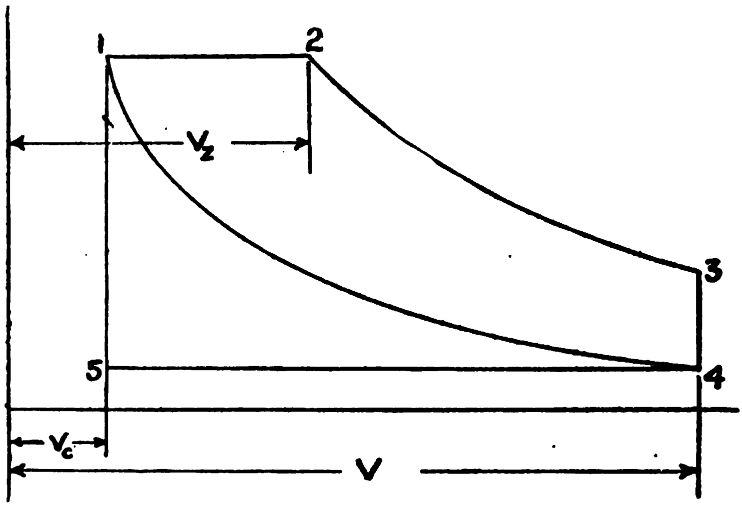
Expressing T_2 in terms of T_1, and T_3 in terms of T_4, we have

$$T_2 = T_1 \frac{V_2}{V_c}$$

and
$$T_3 = T_4 \left(\frac{V_2}{V_c}\right)^x$$

*Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 156.

†Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 160.



If we put $\frac{V_z}{V_c} = \delta$, we will have

$$T_2 = T_1\delta$$

and $$T_3 = T_4\delta^x$$

Substituting the above in the expression for E, we have

$$E = 1 - \frac{K_v}{K_p}\left(\frac{T_4\delta^x - T_4}{T_1\delta - T_1}\right)$$

$$= 1 - \frac{K_v}{K_p}\frac{T_4}{T_1}\left(\frac{\delta^x - 1}{\delta - 1}\right),$$

and since $\frac{K_p}{K_v} = x$, and $\frac{T_4}{T_1} = \left(\frac{V_c}{V}\right)^{x-1} = \frac{1}{r^{x-1}}$

we have finally

$$E = 1 - \frac{1}{r^{x-1}}\frac{1}{x}\left(\frac{\delta^x - 1}{\delta - 1}\right).$$

This equation differs from that obtained for the Otto cycle by the factor $\frac{1}{x}\left(\frac{\delta^x - 1}{\delta - 1}\right)$ in the last term.

A brief inspection will show that with values of δ and x greater than 1, which always holds true in this case, this factor as a whole is greater than one. That means that on purely thermodynamic grounds, with the same degree of com-

pression, the Otto cycle will show a greater cyclic efficiency than the Diesel cycle. The cyclic efficiency of the latter depends not only upon r and x, but also upon δ, and it decreases as δ increases, *i. e.*, as the load on the engine increases. This fact may serve to explain why Diesel engines on numerous tests have shown an increase in the thermal efficiency as the load decreased from the normal.

It may be remarked that the formula is not rigidly correct for this cycle, as we proceeded on the assumption that the weight of charge remains the same throughout, which is not the case. But the increase in the weight of charge for oil fuel is very slight, while it might be considerable for a gas fuel. The only successful machine employing this cycle, the Diesel, is an oil engine, so that this is the only case that need be taken into account. The error made in applying the formula to this case is therefore small, and would tend to somewhat further decrease the efficiency.

For a pressure of 475 pounds absolute at end of compression, and values of x and r as assumed in the table, the following figures show how the efficiency depends upon a variation in δ, *i. e.*, in the load on the engine:

$\delta =$	*1.5*	*2.0*	*2.25*	*2.50*	*2.75*	*3.00*
$x = 1.30, r = 15$............	$E = .532$	.503	.490	.478	.468	.459
$x = 1.40, r = 10$............	$E = .567$	.535	.519	.507	.494	.483

The cyclic efficiency of the Otto cycle, for the same conditions of x and r, would have been constant for all loads at .556 for the first case, and .602 for the second.

The Otto cycle is therefore on theoretical grounds the one to use, provided it can work with the same compression ratio as the Diesel. Unfortunately practical difficulties step in which make some of the theoretical conditions assumed impossible to realize in practice. The most serious difficulty is the fact that pre-ignition of the charge may occur during compression, if the compression is carried too high. The point at which pre-ignition is apt to occur depends upon the composition of the charge, and is further modified by the ac-

tion of the cylinder walls in any particular case. It may be said in general that the higher the contents of the charge in hydrogen the easier will pre-ignition occur, *i. e.*, the lower must the compression be kept for satisfactory working. A glance at the efficiency formula for this cycle will show that this difficulty at once sets a limit to the attainable efficiency. The Diesel cycle is not open to this objection because here air alone is compressed, fuel and air being kept separate until combustion is desired. In fact, the satisfactory working depends here upon a sufficiently high air temperature, and the limit set is only the fact that high pressures cause trouble in mechanical operation.

The conditions being thus outlined, it becomes interesting to inquire to what extent the conclusions drawn on theoretical grounds are thereby modified. To secure a basis of comparison, it may be assumed that the maximum pressures realized in both cycles are the same. Although the maximum pressure maintained in the Otto but momentarily, the machine parts have to be designed for this pressure, and hence the maximum pressure really sets the upper limit for either cycle. Further, under nominal load, the Diesel cycle works with about 10 per cent. filling of the cylinder, and from this the heat supplied each cycle may be computed. The temperature at beginning of compression is taken at 600 degrees absolute. The specific heat at constant volume of the mixture is assumed at $K_v = .19$, that for constant pressure at $K_p = .26$, making the ratio $\frac{K_p}{K_v} = 1.37$, which may be assumed an average case. It is assumed that the Otto cycle for the same maximum pressure receives the same amount of heat as the Diesel, and on this basis a range of efficiencies has been computed. These figures show what each type of cycle can do when each is working under the same maximum pressure and is furnished the same amount of heat, this amount of heat being assumed that necessary to generate the nominal power of the engine under the conditions chosen.

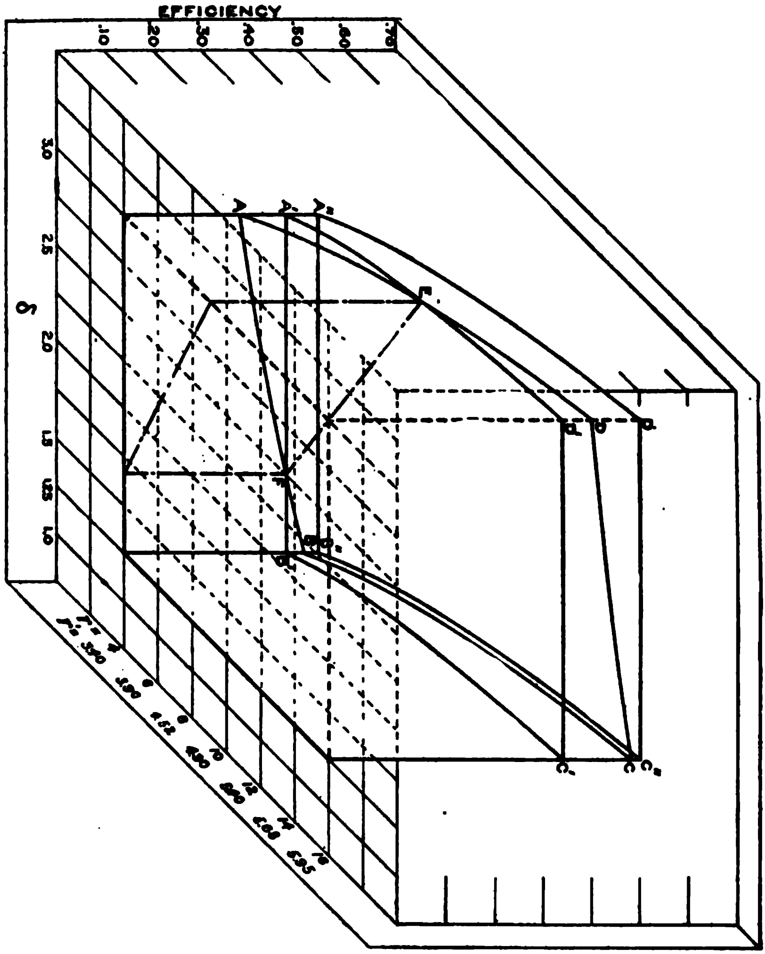
There being three variables present in the formula for the

Diesel cycle, it seems best, in order to give a comprehensive presentation of the field, to use the three co-ordinate system. In the figure shown, distances from right to left present values of δ, from front to back values of r, and from bottom to top values of E*. The methods of constructing the solids is obvious. Three solids are shown. In each case the upper surface shows the variation in efficiency for the ranges chosen. In the case of the Otto cycle the efficiencies are constant from right to left, depending only on r and being independent of δ.

The surface $A\ B\ C\ D$ represents all the efficiencies of the Diesel cycle for values of r from 4 to 16, and of δ from 1.25 to 3.00. In practical working δ is about 2.5, and r varies from about 13 to 16.

The surface $A'\ B'\ C'\ D'$ similarly shows the efficiencies of Otto cycle under the condition above assumed. These two surfaces intersect in the line $E\ F$. For that part of the field lying in the surface $A\ E\ F$, the Otto cycle is more efficient than the Diesel; *i. e.*, assuming $\delta = 2.5$ as the normal condition, the value of r for the Diesel must be about 8 in order to be on a par with the Otto engine. For the rest of the field the Diesel shows a superiority, and since $r = 8$ is lower than what is ordinarily used for this cycle we reach the conclusion that, under normal condition, the Diesel shows a better efficiency than the Otto, which is contrary to what is obtained on purely theoretical ground. It must, however, be remembered that in this discussion no account is taken of probable effect of cylinder walls, and, further, that different assumptions for K_p and K_v will lead to somewhat different results. Güldner, for instance, on the basis of some computations comes to the conclusion that the two cycles when worked under the same maximum pressure show but a slight advantage in favor of the Diesel, so slight that a small increase in the filling of the cylinder above 10 per cent. reverses the results. To realize this the intersection $E\ F$ would have to be much nearer to the diagonal $D'\ B'$. The difference is quite likely

* r' represents the equivalent ratio of compression for the Otto cycle. As an example, if $r = 10$ for the Diesel cycle, the Otto cycle having the same maximum pressure, 352 pounds, and the same amount of heat furnished per cycle, 329 *B. T. U.*, will have a ratio of compression $r' = 4.90$.

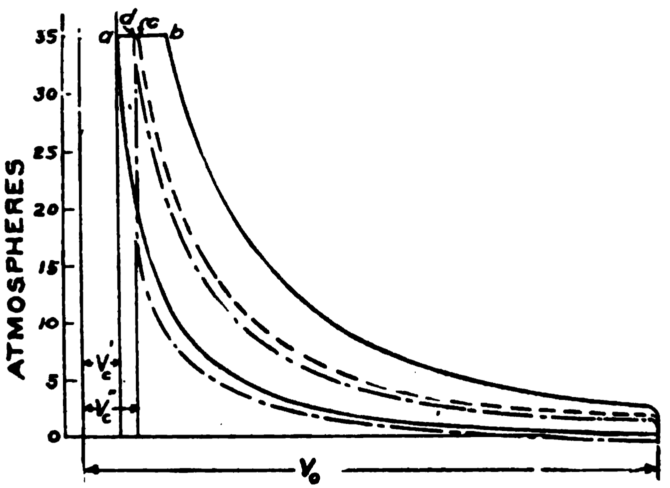


due to a difference in the assumptions. On the basis of his results Güldner is inclined to explain the superiority of the Diesel, as regards thermal efficiency on the indicated horse-power as realized in actual practice, not so much on the ground of superiority of cycle, but on the ground of complete vaporization of fuel and complete and perfect mixture of fuel and air.

To complete the picture, the surface *A'' B'' C'' D''* has been constructed, showing the efficiencies of the Otto cycle under

the same compression ratios as the Diesel. This is, of course, as explained, the ideal case. The efficiencies are throughout higher than those of the Diesel cycle.

Attempts have been made to overcome the difficulty of pre-ignition in the Otto cycle by making the composition of the charge such that pre-ignition is not likely to occur easily. The most successful of these is the device of Professor Donat Banki, who uses a mixture of gas, benzine or petroleum, water and air for the charge.* The heat necessary to vaporize the water keeps the charge during compression at a comparatively low temperature, consequently higher compression pressures can be used without pre-ignition. In the article mentioned the volumes of cylinder and clearance given furnish a value of r equal toabout 10, $\frac{K_p}{K_v}$ is given as 1.4. From these values the cyclic efficiency is about .600. The maximum explosion pressure for full load is given at about 655 pounds absolute.



The compression ratio of the Diesel cycle for this maximum pressure would have been about 16, and for normal load, the cyclic efficiency is about .565. This shows clearly that the superiority of the Diesel cycle may be easily overcome by

* See "Sibley Journal," April, 1903.

slight changes in the conditions. The pressure at the end of compression for the Banki Motor was about 250 pounds absolute, for the Otto cycle, $r' = 6$ in the diagram, it is about 172 pounds absolute. These can not be directly compared with absolute accuracy, since $\frac{K_p}{K_v}$ is not the same in the two cases, but the example shows what a decided change an increase of about 80 pounds in the pressure at end of compression produces.

In conclusion, an ingenious example of how theoretical considerations will sometimes furnish the key to the causes of variations in results obtained on actual tests was given by Prof. E. Meyer, in the "Zeitschrift des Vereins Deutscher Ingenieure," May 9, 1903. Tests on a 14-H.P. Marienfelde alcohol motor, and a 70-H.P. Diesel motor gave the following results:

THERMAL EFFICIENCY ON BRAKE H.P. LOAD.

	Full	*Normal*	¾	½
70-H.P. Diesel motor, per cent............	32.6	31.9	30.5	27.4
14-H.P. Marienfelde motor, per cent.....	32.7	29.0	...	22.7

It will be noted that the efficiency is much better sustained throughout the range of load in the Diesel than in alcohol motor. Meyer gave the explanation for this on theoretical grounds as follows:

Equal friction losses are assumed. In the figure the full line represents the full-load Diesel card, the dashed line the half-load Diesel card, and the dot-and-dash line the card of the alcohol motor. Combustion in the Diesel engine is assumed to be along the constant-pressure line *a b*. The compression ratio $r = \frac{V_0}{V'_c} = 15.40$ for the Diesel, and $r' = \frac{V_0}{V''_c} = 10.26$ for the alcohol motor. Pressure at end of compression is 510 pounds absolute in the Diesel, and 240 pounds absolute in the alcohol engine. Line *a b* is found to be .09 V_0, and line *a c*, for half load, = .04 V_0. A vertical line drawn at V''_c, cuts line *a b* at *d*, inside of point *c*.

Suppose each diagram is divided into a number of infinitesimal elementary diagrams by a number of adiabatics. It will at once be seen that $r' = 10.26$ for each one of the elementary cycles of the diagram from the alcohol engine. From the formula for E, all of these elementary cycles have therefore the same cyclic efficiency. In the case of the Diesel full-load card, all the elementary cycles to the left of d have a ratio of compression greater than 10.26, and these are therefore of higher efficiency than the elementary cycles of the alcohol motor. But all those lying to the right of point d have a value of r constantly decreasing as we pass further to the right toward b, and the cyclic efficiency therefore constantly decreases. The average of the efficiencies obtained for all the elementary cycles of the full-load Diesel card will therefore just about equal the efficiency of the alcohol diagram, because c and d are close together, and hence we find the thermal efficiencies of the two engines actually about equal at full load. For half load, however, the case is different. The alcohol-motor card does not change sensibly owing to the method of regulation. But the average of the elementary Diesel cycle efficiencies is now greater than that of the alcohol-motor cards because but few elementary Diesel cycles lie to the right of d. Hence nearly all of them have a value of r greater than 10.26, and hence the cyclic efficiency is greater on the average. We would therefore expect a higher thermal efficiency at half load from the Diesel than from the alcohol motor, which is what actually occurs as shown on test.

II. FOUR-CYCLE *vs.* TWO-CYCLE ENGINES.

The development of the gas engine, more particularly of the high-powered engine, has made the question of 4- *vs.* 2-cycle engines of the highest importance. The literature on this subject is not at all extensive, and what there is does not seem to bring the controversy to any definite conclusion. The main reason for this state of things is, perhaps, the absence of very conclusive tests, for it cannot be denied that however

much a subject may be discussed theoretically, practical tests, if they can be made, will in all cases decide the question.

The history of the subject is in brief as follows. The four-stroke cycle is undoubtedly the idea of Beau de Rochas, who fully explained its action in a pamphlet of 1862. About the same time Otto attempted to make the 4-cycle engine a practical success. Unable, however, to overcome the difficulty caused by the severe blows of the exploding gas mixtures, he in 1862 gave up the development of the direct-connected 4-cycle engine, and for the next fifteen years, in company with Langen, developed the free piston engine. This engine won them the first prize at the second Paris Exposition. Apparently about 1875 Otto returned to his original idea, and his success along this line was well shown at the Paris Exposition of 1878. Among 75 machines there exhibited, the Otto 4-cycle engine was distinguished above all others both by quietness of operation and by greater power per cubic foot of cylinder volume.

From 1874 to 1886 the Gas Motoren Fabrik Deutz monopolized the construction of 4-cycle engines, under the protection of German Patent No. 532. After a legal fight, lasting nearly four years and going on at the same time in Germany, Austria, England, Belgium, France and Italy, the scope of the patent was considerably reduced, Claim No. 4, covering the 4-cycle principle, the especial bone of contention, being vacated. In the meantime, under the force of circumstances, the other manufacturers were compelled to develop the 2-cycle engine, and there is no doubt that if the state of affairs as outlined had continued to the end of the patent, the 2-cycle engine would have reached its present development much earlier. The very fact, however, that the 2-cycle principal in its infancy offered numerous and serious difficulties, made the legal fight protracted and bitter, the above mentioned pamphlet of Beau de Rochas, of 1862, being a very effective weapon against the validity of Claim No. 4. The natural result of the vacation of this claim in 1886 was that the 2-cycle machine almost disappeared from the market for

nearly 10 years, and has found its present development within the last 6 or 8 years.

As stated at the outset, the question of choice between the 4-cycle and the 2-cycle machine as regards economy, operation and maintenance, is not at all decided, although it must be admitted that the decision commences to lean more and more strongly in favor of the 2-cycle. Güldner, especially, to whom we shall be indebted for a great deal of what follows, decides very strongly in favor of the 2-cycle machine. Other authorities are not so decided. There is a great deal that may be said for and against each type on both theoretical and practical grounds. Before comparing the two methods as to efficiency, it will be well to outline the conditions attending each.

The two types are thermodynamically equal as far as the combustion of the charge, whether at constant volume or constant pressure, and their expansion strokes are concerned. They differ only in their method of displacing the old and taking in the new charge, *i. e.*, in their scavenging and loading.

In the 4-cycle machine the scavenging is done by the power piston pushing out the burned gases through the exhaust port. This is followed by the suction stroke, during which the new charge is taken in. The compression stroke, combustion, and expansion stroke following complete the cycle. The piston therefore works half the time as a power piston, and the remainder of the time as a pump piston.

In the 2-cycle machine the scavenging and loading are done in various ways. In the first place we may distinguish two scavenging agents: the fuel mixture and air alone. Further the scavenging agent is introduced into the cylinder by pumps in various ways. Among these we may distinguish three types: the enclosed-crank case employed as a pump, the front end of the cylinder employed as a pump, and a pump entirely independent of crank case or power cylinder. A further modification comes in when any of the above types of pumps are used with or without an air receiver. Several methods of scavenging are also employed. The fuel mixture alone may

be used to drive out the exhaust gases, or a little air may be sent into the cylinder ahead of the fresh gas, or, finally, the scavenging may be done by an excess of air, followed by the admission of fuel mixture when the scavenging is complete. How these various methods compare among themselves, and with the 4-cycle, will be seen later on.

The final judgment upon the success or non-success of an engine should always be based upon the thermal efficiency of the machine at the crank shaft and upon the mechanical efficiency. Of the power generated during the working strokes of the cycles a part is lost in the fluid friction of the machine and in the rubbing friction of the various machine parts. The fluid friction in a 4-cycle machine will be understood to mean the pump work as indicated by the bottom loops of the indicator cards. In careful testing these loops should always be taken with a weak spring, in order to determine the pump work with more accuracy. This is especially desirable since little data exists on this point. Should the 4-cycle engine be of the positive scavenger type, as the 500-H.P. Premier reported upon by Humphrey in 1900, this pump work should be added to the fluid friction of the bottom loops. In a 2-cycle engine the fluid friction is the pump work done by the fuel and air pumps. The rubbing friction of the machine need not be further defined.

It seems that engineers are not quite unanimous in their methods of computing mechanical efficiency, as the writer had occasion to notice lately in looking over some tests. If from the total indicated horsepower we subtract the fluid friction, in either type of engine, we shall obtain the net indicated horsepower. The mechanical efficiency is, according to some,

$$E = \frac{\text{B.H.P.}}{\text{total I.H.P.}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

according to others,

$$E = \frac{\text{B.H.P.}}{\text{net I.H.P.}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Güldner consistently employs formula (2), which gives a higher result than formula (1), and takes pains to correct

Thurston's figures on a Brayton gas engine, computed according to (1). It seems to the writer that the use of formula (2) is justified only when the work of the air pump is to a great extent recovered in the power cylinder, as is the case in a Brayton engine, or in the Diesel. When the air compressed by the pump is, however, used for scavenging only, blowing through the power cylinder when the exhaust port is open, this is not the case, and for such engines, the majority of 2-cycle engines, it seems nearer right to use formula (1). Similarly the thermal efficiency at the brake should be

$$E' = \frac{\text{Thermal equivalent of B.H.P.}}{\text{Thermal Units in Fuel supplied}}.$$

A complete comparison of the 4-cycle and 2-cycle principles should include the following heads:

I. Thermodynamic actions in the cylinder.
II. Fluid friction.
III. Friction of the machine.
IV. Limitations of construction and economic considerations.

I. THERMODYNAMIC ACTIONS IN THE CYLINDER.

Thermodynamically the methods are equal, the utilization of the heat in practice is, however, influenced by several extra-thermodynamic actions. In the first place, the cylinder volume of most 2-cycle engines is much smaller, approximately about one-half, than the cylinder volume of a 4-cycle machine of the same power. The enveloping cylinder surface is therefore greater per unit volume in the 2-cycle than in the 4-cycle engine. This means that more heat will be carried off by the cooling water during the compression, and the higher compression pressure that this fact permits of carrying means finally an increase in the thermal efficiency. Of course the increased loss of heat also takes place during combustion and expansion, affecting the results unfavorably. It seems, however, that the somewhat lower average temperature of the highly-compressed lean mixtures which may be used in the 2-cycle engine about evened up this loss. As a matter of fact, tests have shown

that the proportionate heat loss to the jacket water does not differ much for the two methods, varying only with size of engine—sometimes in favor of one, sometimes of the other—of the two methods.

Thus it seems that the 2-cycle engine may use a slightly higher compression pressure than the 4-cycle for the same mixture, with its attendant favorable result. This advantage is further increased by the fact, as pointed out by Güldner, that the 2-cycle, if properly handled, works with a lower temperature at the beginning of compression than the 4-cycle, at least in an engine employing air for scavenging, since the scavenging action takes place through only about $\frac{1}{8}$ of the stroke as a maximum, allowing but little time for the taking up of heat from the walls. Further, the scavenging air is usually under a pressure slightly greater than atmospheric, and this and the lower temperature mean finally a greater weight of charge per cycle and consequently greater specific power. In the 4-cycle the loading takes place under suction, lasts during the entire stroke, the charge has had considerable time to warm up, and as a result we have a higher temperature and a lower pressure at beginning of compression than in the 2-cycle machine. Both of these tend to lower the pressure realized at end of compression, the effect of the lower initial pressure being obvious, and that of the higher initial temperature serving to decrease the weight of the charge. Besides, depending of course on the nature of the fuel used, a high initial temperature may set a low limit to final compression pressures because premature explosions must be avoided. Without reference to the effect of temperature, if we assume the pressure of the scavenging air after scavenging to be about 15 pounds absolute, that at the end of the suction stroke in a 4-cycle machine to be about 14.0 pounds, the weight of charge in the 2-cycle will be $\frac{15-14}{14}$ = about 7 per cent. greater than in the 4-cycle machine. We gain about that much in specific power, *i. e.*, power developed per cubic foot of piston displacement, because, other things being equal, the heat converted into external work is directly proportional to the weight of the charge.

Summarizing the above we have the following:

1. Lower temperature and higher pressure of charge at beginning of compression in the 2-cycle, resulting in

2. Greater weight of charge and consequent greater specific power.

3. Greater loss of heat to cooling water during compression, allowing of higher compression pressures, and resulting in

4. Higher thermal efficiency.

Tending to show the correctness of the above discussion, Güldner cites the case of the Diesel-Güldner 2-cycle engine, in which the pressure at end of compression had to be made from 60 to 90 pounds higher than in 4-cycle machines, in order to attain temperatures high enough to ignite the charge.

II. FLUID FRICTION.

The pump work in a 4-cycle machine is done in the power cylinder itself. The disadvantages attending this method have in part been already pointed out. The efficiency of the power cylinder as a pump as compared to an independent pump must be low, because, to state it briefly, the power cylinder is not designed as a pump. The valves are not designed as pump valves, being designed to stand the much higher pressures of the power strokes, and the large clearance spaces and high temperature of cylinder walls affects very seriously the capacity of the cylinder as a pump. It may be objected that the fluid friction in a well-designed 4-cycle machine is not a very serious loss, as may be judged from mechanical efficiencies obtained, but it can not be denied that if the same work were assigned to a cylinder designed as a pump, with proper ports and valves and cool cylinder walls, the same work would be performed at a still lower cost. Besides, as Humphrey pointed out in a test on a Crossley 400-H.P. 4-cycle engine, not only the length, size and arrangement of piping affect the fluid friction very seriously, but the load on the engine also changes it. The amount of this change depends, however, on the kind of fuel used and on the system of governing. Humphrey found that when only air was taken in

the bottom loop cards indicated a fluid loss of about 33 H.P. at normal speed. When exploding every cycle this loss was about 15 H.P. The fluid loss at full load is therefore about $\frac{15}{400} = 3.8$ per cent. Even if we assume that the fluid loss at half load is not at all increased, which can not, however, be fairly expected on the basis of the above figures, the fluid loss would have increased to $\frac{15}{200} = 7.5$ per cent. And with a loss of from 20 to 22 H.P., which is probably nearer the truth, it would have risen to from 10 to 11 per cent.

Finally, unless the engine is of the positive scavenging type, the scavenging in a 4-cycle engine cannot be perfect, since the combustion chamber is always left full of the burned gases. This is unavoidable, and its most serious effect is to decrease the weight of charge in the cylinder, both by its own bulk and by unduly heating the incoming fresh gases by mixing with them.

It is evident from the above that the pump work if performed by the power cylinder can not be done to best advantage. Cutting out the two pump strokes at once converts the machine into a 2-cycle engine, and we have next to examine how the case stands when the pump work is performed in a pump other than the power cylinder. As pointed out above, an examination of existing types of 2-cycle engines shows that this work is done in several ways, some of these ways being undoubtedly responsible for the present bad repute which the 2-cycle machine now enjoys in some quarters. Before deciding between the various means employed it will be well to outline the conditions that attend the scavenging and reloading of the cylinder of a 2-cycle machine.

At the end of the expansion stroke in the 2-cycle machine the cylinder is filled with burned gases at approximately atmospheric pressure. These gases must be replaced to give room for the next charge, and upon the perfection of scavenging depends the capacity, economy and satisfactory working of a 2-cycle engine. This is, in fact, the critical phase of the 2-cycle principle, and it is not too much to say that, depending upon the perfection of scavenging, the 2-cycle will stand or fall.

While the scavenging of the 4-cycle engine is mechanically a simple process, the same cannot be said for the 2-cycle engine. The action going on during the scavenging of a 2-cycle machine is very complex, mainly on account of the time allowed for the process. While the theoretical aspects of the process are very clear, the practical side is not so simple nor so well known. The reason for this is probably that manufacturers of 2-cycle machines are rather backward about giving out their good or bad experiences. Another reason may be, as Güldner somewhat sarcastically remarks, that "to science and to the mechanical laboratories of our technical institutions the 2-cycle principle does not seem to offer any further questions worthy of investigation."

The ideal scavenging process in a 2-cycle engine would be the following: scavenging should commence when the pressure of the burned gases has fallen to nearly atmospheric pressure. In order to insure a rapid establishment of equilibrium after the exhaust port has opened, this port should be a ring of openings in the cylinder wall instead of a valve or opening of the ordinary type. Scavenging should, therefore, commence before the piston has reached its outer dead center. It is of extreme importance, however, and for obvious reasons, that the pressure of the burned gases be less than that of the scavenging agent, and it is better, therefore, to be a little too late than too early with the admission. The scavenging agent should be under low pressure. High pressure causes it to flow into the cylinder under high velocities, besides unnecessarily increasing fluid friction. It is apt to pierce the burned gases, strike the cylinder walls, and rebound from them. This causes eddy currents and sometimes results in a very thorough mixing up of burned and incoming gases, which is just what is to be avoided. If, on the other hand, low pressures be employed, the incoming gases enter quietly, and if the ports have been properly designed, they will fill the cylinder from wall to wall in a solid cylinder, pushing the burned gases out ahead of them. Low pressure and properly-placed inlet ports are therefore essential. This, on the one hand,

again emphasizes the importance of delaying the admission of fresh gas till near the end of the outstroke, and, on the other hand, points out the necessity of careful design of combustion chamber and ports. As a last requirement an excess of the scavenging agent under as nearly as possible constant pressure should be used, for only in this way can thorough scavenging be assured, and this means, indirectly, that scavenging should continue as long as possible, *i. e.*, to the closing of the exhaust ports on the return stroke. It is true that the beginning of compression is thereby delayed until from 10 to 12 per cent. of the return stroke is completed, but the greater weight of charge present in the cylinder, as compared to the 4-cycle, evens up this loss.

Summarizing, we shall therefore have for ideal conditions of scavenging in the 2-cycle engine:

1. Commence scavenging near the dead center on the out stroke, and continue to the closing of the exhaust port in the return stroke.

2. Use scavenging agent under low pressure, high pressure resulting in failure. At present the pressures range from 5 to 10 pounds by gauge. Have properly designed combustion chamber and inlet ports, to prevent as far as possible any breaking up or shattering of the incoming gases. Failure in this will mean imperfect scavenging.

3. Use an excess of the scavenging agent, and maintain as nearly as possible a constant pressure during the operation.

In the light of the above conditions, we are now able to compare the methods of scavenging employed in existing 2-cycle engines.

Referring to the scavenging agents used, and the methods of using them, the engines employing fuel mixture alone are under a great disadvantage because no excess of the scavenger can be used since this would result in a direct loss of fuel through the exhaust port. Imperfect scavenging is the inevitable result, and since it is next to impossible to prevent the loss of some fuel through the exhaust port, such engines can have no claim to economy. Where the crank case is used for

compressing the mixture a further fuel loss is apt to result through leaky frame and bearing joints. Besides this, the crank-case pump is very inefficient owing to large clearance spaces, and it possesses the further disadvantage that scavenging ceases as soon as the piston starts on the return stroke, thus losing a good part of the scavenging period. In view of all these disadvantages we find that this method of scavenging and loading is confined to small 2-cycle machines in which a low first cost is usually of much more importance than great or even fair economy.

If the ready-fuel mixture is sent into the power cylinder by an independent pump, the case is much better as far as pump efficiency is concerned, but the fuel loss through the exhaust port, which is the most important part, still exists.

The above considerations point to the conclusion, first, that a crank-case pump cannot be expected to give satisfactory results, and, second, that, except in such cases where economy is of secondary importance, air should be used for scavenging, either wholly or in part.

Concerning the choice of pumps, an independent air pump with an ample air receiver furnishes the ideal conditions as above outlined, for, under these conditions, we can commence or cease scavenging at will, dependent upon the setting of the valves, and, if the receiver is large enough, an excess of air under fairly constant pressure is available. If the receiver be made too small, scavenging will cease too early, an excess of air not being available, and the scavenging pressure decreases very rapidly. Cutting out the receiver altogether, and discharging from the pump directly into the power cylinder, makes the conditions much less favorable, for, if the main crank and the pump crank are 180 degrees apart, scavenging ceases when the main piston has reached its outer dead center, and if the pump crank be made to lag or lead a few degrees, as the case may be, the scavenging pressures are apt to be unduly high, and they decrease very fast. In the third type of pump, employing the front end of the power cylinder, we have two other points entering. One is that no excess of air

is available, the other, that the capacity of the pump is further decreased by the unavoidable heating of the air during suction by contact with the cylinder walls. These considerations lead to the following ranking of the types of pumps in order of merit.

1. Independent pump with ample receiver.
2. Independent pump without receiver.
3. One end of power cylinder employed as pump cylinder.
4. Crank-case pump.

Whether air entirely or only in part be used for scavenging again depends upon consideration of first cost mainly. The idea of using air in part only is that, if a little air is sent into the cylinder ahead of the fuel mixture, if any of the fresh gases are lost through the exhaust ports, it will be the air alone. This has in some cases been accomplished by letting the fuel pump suck air only during the latter part of its stroke. The air will then be the first to enter the power cylinder. This method cuts out the air pump, and has this fact in its favor. In other cases, as in the Koerting, the air pump fills the pipes leading from the gas pump to the cylinder with air, so that as the gas pump starts to deliver, it first presses this air into the power cylinder. The success in either case depends upon the assumption that the three layers of gas at one time in the power cylinder, *i. e.*, burned gas, air and fuel mixture, remain separate during the scavenging. Any production of eddy currents in the cylinder will defeat the design, and it can at once be seen that failure is much more likely to result than success. The use of air alone for this work therefore seems to offer the best chance of accomplishing the desired end.

The following cards taken from air receivers and crank cases show the action in each during scavenging.*

Fig. 1 is taken from an air receiver which was too small. The pump delivers air into the receiver from *a* to *b*. The maximum pressure is about 5 pounds, and this remains constant to *c*. At *c* scavenging commences, and the pressure falls

*Güldner Entwerfen und Berechnen der Verbrennungsmotoren, pp. 186, 187.

rapidly to about 1.5 pounds at *d*, near the end of the power piston stroke; after that very little air flows from the receiver, and the best part of the scavenging stroke, from *d* to the closing of the exhaust port, has not been made use of.

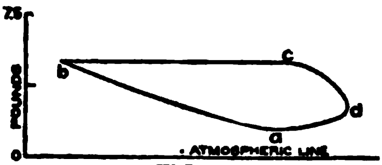


FIG. 1

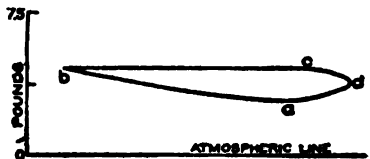


FIG. 2

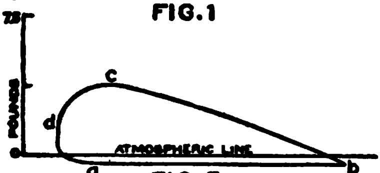


FIG. 3

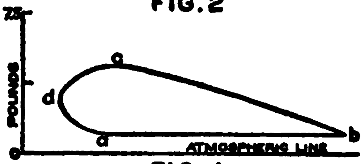


FIG. 4

Fig. 2 shows a card taken when the receiver volume was considerably increased. Here the maximum pressure is not quite so high, but scavenging is continued from *c* through *d* to *a*. The pressure drops only to about 3 pounds, and since this is the pressure also existing in the cylinder at the close of the exhaust and air ports, more weight of charge is present in the cylinder, increasing the capacity of the machine.

Fig. 3 shows what takes place in the crank case. Suction occurs from *a* to *b*, compression from *b* to *c*, at *c* air is admitted to the power cylinder, and the pressure rapidly drops to *d*. Here the return stroke commences, scavenging ceases, and the pressure rapidly drops to suction pressure.

In Fig. 4 the conditions of Fig. 3 have been improved by making the pipe leading from the crank case to the cylinder as large as the space would allow and fitting this pipe with a check valve, thus creating a limited receiver. This bettered the conditions very much, scavenging taking place from *c* through *d* to *a*, continuing after the piston has commenced its return stroke, owing to the action of the check valve. Only in this way is it possible in crank-case scavenging pumps to attain anything like favorable results.

From the foregoing it is clear that if the 2-cycle engine is

to compete successfully with a 4-cycle machine, the scavenging must be done carefully and thoroughly, and, in order to do this, independent pumps and ample air receivers are essential. Every other means is but a compromise, and should come under consideration only where first cost is the main factor, and economy but of secondary importance. Further, it would seem on the purely theoretical grounds so far outlined, that to do this work should cost less in the 2-cycle than in the 4-cycle machine. It is not easy to show whether this is true in practice or not from existing data. Recorded tests usually are incomplete in that they give the difference between indicated and developed horsepower only without differentiating the friction loss into fluid friction and friction of the machine. Güldner feels nearly sure that the fluid loss in the 2-cycle can be brought down to or be made less than that in the 4-cycle. According to him, this loss in the 2-cycle machine is at present about 8 to 10 per cent. of the indicated work, as compared to from 6 to 7 per cent. in the 4-cycle, but he feels very certain that as the 2-cycle type is improved this difference of from 2 to 3 per cent. will ultimately disappear. Schöttler, in the "Zeitschrift des Vereins Deutcher Ingenieure," Oct. 4, 1902, speaking of the engines exhibited at Düsseldorf in 1902, says: "The 2-cycle has the unquestioned advantage that but half the cylinders are required (for same power). This is negatived to some extent by the fact that pumps are necessary, but these will take more power than the pump work in the 4-cycle." No figures are given to support this statement, but Güldner agrees with him in so far as he admits that the 4-cycle has a slight advantage today in respect to fluid friction. This, however, may merely show that the practice of today has not been able to fully realize the advantages offered by the 2-cycle machine, which is not at all surprising when the period of development is considered. The conclusion would be that the difference, if any, is as yet in favor of the 4-cycle as regards fluid friction, but that as experience teaches and practice improves, this difference will disappear.

III. FRICTION OF THE MACHINE.

Turning next to the friction of the machine, the points to be considered here are the friction of pistons, crossheads and bearings. If we consider that the rubbing friction in an engine is in direct proportion to the volume of the cylinder per unit of power, then, if the pump volumes are added to the cylinder volume in the 2-cycle, the two methods would seem to be about on a par. The pump piston in a 2-cycle machine, however, works under comparatively low pressures, and the rubbing surfaces are cool and well lubricated. The pressures being low, the piston rings may be given an easy fit, and, all these things considered, the friction in the pump should be low. In the 4-cycle engine we have a machine member, the piston, designed to stand pressures of from 250 to 450 pounds, working half the time under pressures of from 14 to 16 pounds, a state of things not conducive to best results. Under such circumstances it is hardly necessary to state on which side the advantage as to piston friction lies. Nearly all of the smaller-size motors use a trunk piston, working without a crosshead. This is done in both 2-cycle and 4-cycle machines. The practice can not be called a good one, because it makes the work of the piston very severe, using highly heated rubbing surfaces to take up transverse stresses. If a crosshead is used, there is no good reason why a difference in the rubbing friction in this member should exist when employed in either type of machine. In the 2-cycle machine a crosshead possesses the double advantage that it relieves the working piston from other than longitudinal stresses, and it may be designed to serve as a piston for the pump. In regard to bearing friction, the 2-cycle machine, with a smaller piston for the same capacity, has somewhat smaller dimensions in bearings and journals, and hence the friction loss is somewhat less at these points. The fact that an impulse is given every revolution also allows of a smaller weight of fly wheel for the same requirements of regulation, and this, reducing the pressure on the bearings, of course serves to further decrease the friction loss. All of this finally seems to point to a lower total friction loss in the 2-cycle engine.

As already pointed out, the conclusion under II and III cannot at the present time be substantiated from the existing data. Of 4-cycle tests there are a great many, but the number of 2-cycle tests is very small. The practical proof is, of course, a demonstration of equality as regards mechanical efficiency. This proof can not at present be brought. The writer has collected all the available tests from a great variety of sources, but while they show that the *best* figures are obtained by the 4-cycle engine, it is by no means clear that the *average* figures for the 4-cycle are better than those for the 2-cycle.

IV. LIMITATIONS OF CONSTRUCTION AND ECONOMIC CONSIDERATION.

While the considerations under II and III seem to point out that the 2-cycle is but little behind the 4-cycle engine as regards power losses in the machine, and that what little advantage the 4-cycle now possesses will probably disappear as soon as the conditions governing fluid loss are better understood, it must not be concluded that it is therefore immaterial in any given case whether a 2- or a 4-cycle machine be used. Questions of construction and of economy enter the problem, and these will usually have an important bearing upon the final choice.

According to speed and power, gas engines may be roughly divided into the following three classes:

1. Speeds in the neighborhood and above 500 r.p.m., power comparatively low, not to exceed 20 to 30 H.P., as a maximum. Such engines are used in the automobile industry, for marine launches, etc.

2. Speeds from 200 to 500 r.p.m., power up to about 100 H.P. This class includes the engines used for general all-round power work and production of electricity.

3. Speeds below 200 r.p.m., and power up to 500 H.P. and over. These engines have as yet limited application, being used mainly in connection with blast-furnace gas, for the generation of current and production of blast for the furnaces.

1. The engines used for automobile work are nearly all 4-cycle machines. The same may be said of launch engines,

although many 2-cycle machines are also used for marine work. The use of 4-cycle machines for these purposes is easily explained on several grounds. In the first place economy is of secondary importance in nearly all of these cases. The first consideration, except perhaps for marine work, is lightness combined with power. This at first sight would lead to the choice of a 2-cycle machine, as having a higher power per cubic foot of cylinder volume. The high speeds found, however, in this work make a successful operation of the 2-cycle principle very uncertain because the time allowed for the various events is exceedingly small. The development has therefore been along the lines of 4-cycle machines with very high speeds, to attain power combined with lightness. The results attained are quite remarkable, especially as regards motor for air ships. The firms mainly concerned in this are Daimler, DeDion and Bouton, and Buchet.* In the last half of the 90's, Daimler built for Dr. Wölfert a 6-H.P. motor weighing about 90 pounds per H.P. This was a 4-cylinder machine making 800 r.p.m. Daimler also built the motor used by Von Zeppelin in his remarkable work. Two motors were used, each of 12 nominal H.P. at 700 r.p.m., which could be increased to 16 H.P. for a short time. These machines were an improvement over that used by Wölfert. The weight of each, including cooling water, benzine and accessories, was about 1,000 pounds, which, at 16-H.P. would be about 64 pounds per H.P. Since that time DeDion and Bouton have, by increasing the revolution to about 2,000 per minute and by the use of valves of large cross-section, reduced these figures considerably, although, of course, at the expense of economy. Buchet worked along the same lines. For speeds of about 2,000 r.p.m. Buchet gives the following figures for his automobile machines, the weights are exclusive of vaporizer, benzine, and cooling water :

Power.	No. of Cylinders.	Total Weight.	Weight per H.P.
8 H.P.	2	114 pounds.	14
16 H.P.	4	202 pounds.	13
24 H.P.	4	238 pounds.	10

*Von Paller, Z. d. I., Aug. 16, 1902.

Both Santos-Dumont, in Paris, and Kress, in Vienna, used Buchet motors in their later experiments. Although figures are not available, the writer has been credibly informed that the above figures have been reduced.

2. The choice of the type of machine for general power work is somewhat dependent upon the purpose for which it is to be used. The factors entering are not only economy of fuel, but also first cost, maintenance, certainty of operation and regulation. Admitting that the 2-cycle machine can not at present quite show the economy of the 4-cycle engine, it has the advantage in all of the other items named, with the possible exception of certainty of operation. It may be said, however, in regard to this latter point, that unreliability of operation was associated with gas engines in general in the minds of the public for a long time. Later on the charge was made against 2-cycle machines only. In neither case is it at the present time justified, and in no case except in small very high-speed machines, for reasons above pointed out, does the 2-cycle machine give any trouble in this regard. If the engine is to be employed for the driving of generators, the regulation is of the highest importance. Two-cycle machines have here an unquestioned advantage because of much more uniform crank effort with a lighter fly wheel. The same thing can be obtained by using two cylinders in the 4-cycle type. This, however, camplicates the machine, and merely emphasizes the advantage of the 2-cycle as to first cost and maintenance.

3. Schöttler, in the "Zeitschrift" for June 14, 1902, says, with reference to the engines exhibited at Düsseldorf:

"To make an attempt at a decision as to which one of the two working cycles, now employed in large gas engines, belongs the future, or, to put it more modestly, from which one of the types of engines exhibited will develop the type of the future, would be too early. They all have their advantages and their faults. That the theoretical man leans toward the 2-cycle, and the practical man toward the 4-cycle, is easily understood. It was quite natural, when the iron and

steel industry demanded large gas engines, that an attempt should first have been made to adapt the tried 4-cycle principle. . . . But that the 2-cycle should soon be tried was also not to be wondered at since smaller dimensions could probably be attained. Theoretically it is immaterial as regards regulation whether four 4-cycle cylinders or one double-acting 2-cycle cylinder be used; in either case lighter fly wheels may be used than if a smaller number of cylinders and single action had been employed."

Schöttler also gives the following table of the details of some large gas engines in the same article.

	Brake H.P.	Cylinder. Diameter, inches.	Cylinder. Stroke, inches.	Revolutions per minute.	Piston speed, feet per minute.	Piston displacement for 1 H.P., cu. ft.	Weight, tons, metric. With fly wheel.	Weight, tons, metric. Without fly wheel.	Coef. of regulation.
Single-cylinder 4-cycle Société Cockerill........	575	51.2	55.2	93	860	.115	127	94	1 : 20
4-cylinder 4-cycle Deutz.	600	26.0	33.6	150	840	.068	106	88	1 : 130
Single-cylinder double-acting 2-cycle Körting.	500	25.0	43.4	100	724	.025	70	58	1 : 80

H. Neuman, in a discussion on Meyer's report on a test of a Körting engine, stated that the Deutz company were not building 2-cycle engines, because computations had shown, allowing an indicated pump resistance of 6 pounds, that the mechanical efficiency of a 2-cycle machine was about 74 per cent., while that of a 4-cycle, under the same conditions, was nearly 90 per cent. The economy, everything else being the same, would be in the ratio of the efficiencies. That is a strong statement against the 2-cycle machine, and it would be interesting to know whether practical experience has corroborated these computations. So far nothing conclusive has been published to prove the point.

Agreeing with Schöttler that it is theoretically immaterial

as regards regulation whether four 4-cycle cylinders are employed, or one double-acting 2-cycle cylinder, it is not quite so immaterial in many other respects. Comparing the Deutz and the Körting engines, from the above table we find the piston displacement for B.H.P. in the Körting 2-cycle machine is only 37 per cent. of that in the Deutz 4-cycle engine. The weights per H.P. compare as follows:

	Deutz	*Körting*	*Per cent.* $\frac{\textit{Körting}}{\textit{Deutz}}$
Gross weight per H.P., tons..........	.17	.14	82
Without fly wheel, weight per H.P.	.15	.12	80

This shows a gain of about 20 per cent. in favor of the 2-cycle engine. This, combined with the fact that four cylinders must make an engine much more complicated and more difficult to handle than a single cylinder with pumps, gives a very strong argument for the 2-cycle in large engines.

The co-efficient of regulation, *i. e.*, $\frac{\text{max. speed} - \text{min. speed}}{\text{average speed}}$ is better in the Deutz engine, but it is bought at the expense of a fly wheel 50 per cent. heavier. This not only serves to increase first cost, but tends to decrease mechanical efficiency. For ordinary purposes the co-efficient $\frac{1}{80}$ is sufficiently low to come within specifications.

In conclusion, there is one aspect of the case which grows in importance as the demand for greater power from least expenditure becomes more imperative. More than 500 H.P. have been produced in a single 4-cycle cylinder, but beyond that the difficulty increases very seriously. The large dimensions necessary make very costly construction, and the difficulty of obtaining a proper mixture of fuel and air in such large cylinders, combined with consequent uncertainty of operation and loss of efficiency, soon sets a practical limit to the power generated in one cylinder. For very large machines it is therefore no longer a question of economy only, but a question of possibility of construction and of operation, and, unless recourse be had to complicated multi-cylinder machines, the solution is obvious.

III. MATTERS OF DESIGN, AND THE PRINCIPAL DIMENSIONS.

It may be stated with some satisfaction that the era of hunting after novel constructions, *i. e.*, something not before accomplished, is about passed in gas engine design, and that the subject is being gradually but surely placed on a common sense business footing. Time was when nearly every independent designer was bound to produce something distinctively his own, and this tendency led to many curious and sometimes absurd constructions. Happily it is becoming more and more widely recognized that the underlying principle that should guide all design, *i. e.*, best adaptation of the means at hand to the particular circumstances of every different case, applies with as great force to the subject of gas engine design as it does to the other fields of the mechanical engineer's activity.

In gaining an idea of the true economy of an engine, we must take into consideration all of the costs of operation. These not only include the cost of fuel and of maintenance, but also interest on capital invested, and depreciation. The capital invested depends directly upon the market price, and this in turn upon the manufacturing cost of the machines. In many cases the interest on the invested capital is a large part of the yearly operating cost, and for this reason, and on account of the fact that a low market price will increase the selling qualities of the machine, the designer must always keep in mind manufacturing costs.

It is hardly necessary to point out that the various types of motors should each have its separate treatment in design. The requirements of a marine motor differ from those of an automobile engine, and, in general, the design of small motors should be governed by somewhat different views than that of a large engine. Broadly speaking, a small engine should be designed with a view to the greatest possible simplicity, a large engine with a view to obtaining the highest possible economy. These should be in each case the leading ideas. In the first case, however, economy must not be entirely lost sight of, nor simplicity of construction in the second. The possession of proper judgment as to how far the primary considerations may

be deviated from in favor of the secondary, makes the sucessful designer. One thing must never be forgotten. In all cases reliability of operation and durability are of the utmost importance. One hour of interruption in an important manufacturing plant may swallow up all the saving in fuel of a year, and an engine with the very highest thermal efficiency is but a poor investment if it has to be scrapped at the end of a few years.

There are several minor points which should also receive due consideration. The maximum pressures occurring in gas engines are from 3 to 4 times those occurring in steam engines, and the time of their occurrence is not exactly the same, turn after turn. The engine should therefore be so designed that the stresses may be taken up centrally as nearly as possible. For this reason the Tangye type of frame has not found much application in gas engine design. The reason is that the crank transmits the pressure through a moment arm to the bearing, producing a bending moment on the frame, which, in view of the high pressures occurring, is very severe. Again, one bearing is required to carry the total pressure. Both of these things require a much heavier frame of this type in the case of the gas engine than in that of the steam engine. While steam engine designers have found this frame well adapted to even the highest powers, the altered conditions of gas-engine practice do not allow of its use, at least for large engines. Design the frame so that all stresses may be easily computed and checked. This will in itself lead to simplicity of form, which is conducive to safety and economy, and while this is to be commended it is well to guard against the other extreme, plainness degenerating into ugliness.

The whole arrangement of engine and accessories should be as simple as possible. A complicated layout of valves, cocks and handles merely tends to confusion. A printed set of instructions, if at all elaborate, usually fails to fulfill its purpose, being either never read, or, if read, usually misunderstood. Parts of engines which are set at the factory, and which should not, or can not, be moved by the engine tender without en-

dangering the safety of the machine, had best be fixed rigidly in position, as many attendants have an inquisitive turn of mind.

Among the points of design still discussed in the literature on the subjectare the following: Vertical or horizontal engines, the use of the crosshead, single- or double-acting cylinders, multicylinder construction, and compounding. The views for and against these matters will be given as briefly as possible.*

The gas engine of today, outside of the small machines, is of the horizontal type, the most notable exception being the Westinghouse in some of its forms. It seems to be assumed that the horizontal is the better form, but the vertical arrangement possesses strong points in its favor. In a vertical machine the wear on the piston is much more evenly distributed, and much less, than in horizontal machines. This, with the fact that lubrication is apt to be more uniform, reduces piston friction to a minimum. It is also possible with proper design to nearly avoid bending moments in the frame, leading to a lighter frame construction. This is important, and explains why Diesel and Bánki adhere to the vertical form. In this form it is easy to multiply the cylinder to attain the highest powers, and since no crossheads need necessarily be used, the machines will not need excessive head room. The foundation under this form is subject to vertical stresses mainly, and can therefore be built smaller and at lower cost than that of a horizontal machine of the same power. All of the valve gearing is above ground, and not partly in pits, as is the case in some horizontal machines. Further, the vertical form allows of nearly free expansion by heat in an axial direction, and when it is considered that heads and pistons can be removed with great ease, it must be admitted that the claims of the vertical machine are very strong.

There are, of course, cases where the vertical machine is out of place. Where blast furnace gas, or any of the industrial gases, are used, dust is apt to cause trouble by lodging on the walls or the piston in vertical machines. In horizontal

*For a very complete discussion see Güldner, Die Verbrennungsmotoren.

machines valve ports and combustion chamber may be so designed that the dust will be swept out by the gases. It is also hard to make a vertical machine double acting on account of difficulty experienced in placing and operating the valves in the lower head. Further objections made to the vertical machine are that the valve gearing is apt to be complex and that the crank shaft is not well under control for oiling and adjustment. The former is merely a matter of proper design, and the latter falls when we consider the successful operation of our large marine engines, which are subject to runs of long duration under conditions much more severe than are found in stationary practice.

Whether or not a crosshead should be employed depends upon the size and type of engine. The trunk piston combines two very important functions. On the one hand it must take up the pressures generated in the cylinder and transmit them without any loss through leakage of gases by the piston. On the other hand it is required to take up the oblique stresses caused by the angularity of the rod. It is nearly always bad to make one machine member serve in two capacities, and when the functions so combined are as dissimilar as they are in this case, trouble may result. Loss of pressure by leakage is prevented by the piston rings, and in order to reduce friction to a minimum and to increase durability, they should have an easy fit and run on well-lubricated surfaces. In its function as a crosshead the trunk piston is during the expansion stroke forced against one side of the cylinder, producing a tendency to displace the rings and to cause unequal wear. It is essential that the unit pressure on the side of the cylinder be kept low enough to insure proper oiling and minimum wear. In the earlier years, when the maximum pressures did not exceed from 150 to 180 pounds, and the mean pressures were 50 to 60 pounds, per square inch, it was quite easy to make the piston long enough to insure low unit pressures. And as a matter of fact, in the small and medium-sized machines of today, pressures and dimensions are such that no trouble

whatever need be experienced from this source. The rings may tend to blow through and wear out a little sooner, but this is overbalanced by the great saving in first cost when the crosshead is dispensed with. With maximum pressures from 400 to 600 pounds, and mean pressures in the neighborhood of 100 pounds, the case is, however, somewhat different in large machines. The great pressures produced by the angularity of the rod and the inertia of the great masses of metal require very large trunk pistons in order to keep unit pressures within the desired limit. The inability to properly lubricate large piston surfaces, in extreme cases 50 to 70 square feet, and consequent excessive wear, soon sets a limit to the possibility for constructing engines over a certain power without a crosshead. The disadvantages are greater first cost of machine, greater length, and perhaps a small loss in mechanical efficiency. All of these objections are, of course, overruled by necessity in the case of very large machines, and in such cases the increased first cost and greater length of machines is of secondary importance, while the loss in efficiency is very small when both piston and crosshead are designed to perform their functions properly.

The earliest gas engines were double acting. Their economy, however, was so low that single-acting cylinders were soon employed in order to allow the air to cool the piston. Single-acting machines seem to be the rule today, and for small and medium-sized machines this construction is best, being cheapest and simplest. It is found that, if higher power is desired in these machines, it is cheaper and better to use two single-acting cylinders than to make the machine double-acting. In large machines the case is again somewhat different. All considerations compel the generation of maximum power from minimum cylinder volume. This means the use of double-acting cylinders. Water cooling will have to be employed for the pistons, but since this is necessary also for large trunk pistons, the inconvenience involved should not be charged solely against the double-acting machine. The serious difficulty of this type is in the successful design and

construction of the cylinder head of the crank end. The question of the stuffing box has been successfully solved since the invention of good metallic packings. But the water-cooled stuffing box leaves little room for the valves, and the many openings in the head are bound to weaken it. The idea presents itself to make the piston operate some of the valves in order to remove them from the head, and this leads to the double-acting 2-cycle engine as exemplified by the Koerting engine.

The main reason for employing several cylinders in steam-engine design is to reduce cylinder condensation to a minimum, and thus to save fuel. This reason does not exist in the case of gas engines, but in its place there are several other very strong ones, and among them may be mentioned the following:

1. In single-cylinder 4-cycle machines crank shaft and bearings have to be designed to stand the maximum pressure, although this occurs but momentarily every fourth stroke. This means an extra expenditure of metal and an increased loss by friction.

2. In order to attain anything like satisfactory regulation the revolving masses have to be made heavy. For the lowest coefficient of regulation $\delta = \frac{1}{20}$, which ordinarily occurs, Güldner has computed that a horizontal 4-cycle machine will need about 110 pounds of metal in the rim of the wheel, per horse power, the peripheral velocity of the rim being about 65 feet per second. For a 500-H.P. machine this would mean a rim weighing about 55,000 pounds. The figures are even more striking when we consider the cases of electric lighting and of the operation of alternators in parallel. For the former case, $\delta = \frac{1}{75}$, everything else being the same, the rim would weigh about 187,000 pounds, and in the latter, $\delta = \frac{1}{125}$, the weight would be not less than 330,000 pounds.

3. The manufacturing costs of the parts of large engines will be very high because special machinery is necessary. These in themselves are uneconomical because comparatively little used. To this must be added the uncertainty of successfully obtaining safe and sound machine parts of large dimen-

sions. Both of these things tend to increase manufacturing costs, besides making time of delivery uncertain.

4. Large castings, if at all complicated, give trouble owing to the formation of internal stresses. This disadvantage is augmented by unequal expansion by heat when in service. Unless carefully designed with full allowance for casting stresses and expansion, failure is likely to result, and has resulted in many instances. To these disadvantages must be added the difficulty of properly fitting such large pistons to prevent pressure losses and still have them loose enough to cause no great loss by friction, and the difficulty of properly oiling the piston surface.

5. The heat abstraction by the jacket water grows proportionately less as the cylinder volume increases. In order to prevent premature explosions, the compression pressure must be constantly reduced, which is accomplished, of course, by increasing the clearance. Now large engines nearly always use lean gas mixtures, in themselves difficult to ignite, and the increasing clearance spaces only add to the difficulty. Hence we find that engines generating from 50 to 100 H.P. in one cylinder show the best thermal efficiencies. Up to 150 H.P. there is no great increase in the efficiency, and beyond that there is a decrease rather than a gain.

The above considerations show clearly why above a certain rating more than one cylinder should be employed, this certain rating depending somewhat on other conditions. The main criterion seems to be the requirement regarding the co-efficient of regulations. For general power service one can go much higher with a single cylinder than for electric-lighting service, for instance, without seriously encountering any of the difficulties mentioned. For 4-cycle engines with any of the industrial gases the present ordinary limit seems to be about 300 H.P. in a single cylinder. For stationary engines of other types single cylinders may be employed up to from 50 to 60 H.P. The lower limit of division, however, depends somewhat on the individual circumstances of every case.

The methods of combining the cylinders are various. A

No.	Arrangement of cylinders and cranks.	No. of cylinders.	4-cycle. Angular movement of crank bet. explosions. Degrees.	4-cycle. Weight of fly wheel for Equal dimensions. x_1	4-cycle. Weight of fly wheel for Equal power x_2	4-cycle. I.H.P. of combination.	4-cycle. Weight of flywheel per I.H.P. Pounds.	2-cycle. Angular movement of crank bet. explosions. Degrees.	2-cycle. Weight of fly wheel for Equal dimensions (Sing. cyl. x_3	2-cycle. Weight of fly wheel for Equal power 4—cycle=1). x_4	2-cycle. I.H.P. of combination.	2-cycle. Weight of fly wheel, per I.H.P. Pounds.
I	Single cylinder, single-acting.........	1	720	1.000	1.000	65.8	147.0	360	.802	.401	131.6	59.6
II	Single cylinder, double-acting.......	1	180 & 540	1.230	.615	131.6	91.0	180	.424	.106	263.2	15.6
III	2 cylinder tandem, single-acting....	2	360	.769	.398	131.6	59.0	360	1.595	.399	263.2	59.2
IV	2 cylinder opposed, 1 crank, single-acting....................................	2	180 & 540	1.290	.645	131.6	96.0	180	.335	.084	263.2	12.5
V	2 parallel cylinders, cranks together, single-acting	2	360	.792	.396	131.6	58.0	360	1.602	.401	263.2	54.6
VI	2 parallel cylinders, cranks 180° apart, single-acting..........	2	180 & 540	1.290	.645	131.6	96.0	180	.335	.084	263.2	12.5
VII	3 parallel cylinders, outside cranks together, inside 180° from other, single-acting..................................	3	240	.678	.226	197.4	33.7	120	.237	.039	394.8	5.9
VIII	4 parallel cylinders, cranks 1 and 3 together, cranks 2 and 4 together, and 180° from 1 and 2, sing.-act'g..	4	180	.187	.048	263.2	7.1					
IX	4 cylinders, 2 pairs opposed, 2 cranks together, single-acting.....	4	180	.187	.187	263.2	7.1					

For a 2-cylinder opposed 2-crank engine, use values of Type V. Degrees between explosions = 360.
For the 4-cylinder, 2 pair opposed, 4-crank engine, use values of Type VIII or IX. Degrees between explosions = 180.

very good idea of the comparative merits of the different combinations may be gained from the preceding table, transposed from Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 328. The table is based on computations made on a Koerting 4-cycle engine using illuminating gas. The dimensions were 15 inches by 26 inches, 140 r.p.m. The weight of the fly wheel of this engine is taken as 1,000, and the table shows under

X_1 the relative weights of fly wheels of the various 4-cycle combinations when the cylinder dimensions are unchanged.

X_2 the same for 4-cycle combinations of equal horsepower.

X_3 the relative weights of fly wheels of 2-cycle combinations, as compared with the single cylinder 4-cycle, the cylinder dimensions being unchanged.

X_4 the same for 2-cycle combination of equal horsepower.

The coefficient of regulation δ is taken at $\frac{1}{40}$ in each case, and the velocity of the rim at 65.6 feet per second. The factors obtained therefore apply to an actual case, but they will serve with sufficient accuracy as a guide to the saving effected in fly-wheel weight in general cases.

The columns best compared are X_2 and X_4. They show, for instance, that if a certain horsepower is to be generated in one 4-cycle cylinder, and we call the weight of fly wheel required for $\delta = \frac{1}{40}$ equal to 1.0, we can do the same work in a double-acting 4-cycle with 61.5 per cent. of the fly-wheel weight, in a single-acting 4-cycle tandem with 39.8 per cent. of the fly-wheel weight, in a 3 cylinder single-acting 4-cycle of Type VII with but 22.6 per cent. of the original fly-wheel weight, and, if Type VII be made 2-cycle, but 3.9 per cent of the original weight is required.

In this connection Münzel in Stahl and Eisen, 1900, p. 528, gives a very interesting comparison of the first cost of *blast-furnace gas engines of 1,000-H.P., using various values for the coefficient of regulation, and various cylinder combinations.

*See also Güldner, p. 201.

RELATIVE FIRST COST OF 1000-H.P. BLAST-FURNACE GAS ENGINES.

Required regulation.	Single cylinder	2-cylinder parallel.	Deutz 2-cylinder opposed.	Deutz 4-cylinder opposed.
Ordinary operation, $\delta = \frac{1}{25}$	1	1.05	.90	.95
Electric lighting, $\delta = \frac{1}{76}$	1	.90	.85	.75
A-C operation, $\delta = \frac{1}{125}$	1	.75	.75	.60

Attempts have been made from time to time to improve the thermal efficiency of the gas engine either by obtaining complete expansion in one cylinder or by compounding. In either case the idea is to take advantage of the pressure still existing at the end of expansion in the cylinder in the ordinary case. The first method promises more gain than the second, but it can be shown in either case that the gain amounts to little or nothing. Complete expansion in the cylinder has been obtained by making the suction stroke shorter than the expansion stroke, *i. e.*, by closing the inlet ports before the suction stroke has been completed. This furnishes a very effective method of speed regulation, but fails of its real purpose, for at rated load the suction stroke is nearly as long as the expansion stroke and we revert to the original condition. At any lower load than the rated, the compression pressure is less and less as the load decreases, any gain in efficiency by more complete expansion will soon be negatived by a loss due to lower compression. The balance between gain on the one hand and loss on the other is reached sooner than indicated on account of other circumstances. The longer expansion makes the mean effective pressure less, correspondingly decreasing the specific power. This means larger engine dimensions for the same power, from which it follows that the friction losses are greater proportionately than they were before. To gain anything therefore the compression pressure should be kept constant and the complete expansion be made possible by other means than throttling the suction. The gain will be the greater the lower the original compression, the richer the charge, the greater

the degree of after burning, and the smaller the original friction losses in the machine.

The idea of compounding to gain complete expansion promises no gain whatever, although inventors have made attempts in this direction many times. The reasons for the failure are to be found in the high temperatures and low specific heats of the expanding gases. In order to obtain any reasonable mean effective pressure in the second cylinder, it is usually necessary to exhaust from the high-pressure cylinder at higher pressure than would ordinarily be done. The effect on the high-pressure cylinder is a higher mean temperature, and consequently greater proportionate loss to the jacket water, not to mention greater difficulty in proper lubrication. The very fact, however, that a greater initial pressure is thus obtained in the second cylinder has the great disadvantage that the gases flow over at a much higher temperature. In many cases the after-burning has not even been completed. In any event the valves and ports between the cylinders must be water cooled, if they are to last any length of time, and here is where we find the greatest loss. The abstraction of heat by the necessary water cooling, and the flüid friction in the ports causes so great a drop in pressure that the useful work in the second cylinder is very seriously reduced. These disadvantages seem to be inherent. As a consequence we find that compound gas engines already constructed have shown an efficiency about equal to that of a second-grade single-cylinder engine, and if any gain is to result from complete expansion, it is to be obtained rather in one cylinder than by compounding.

THE PRINCIPAL DIMENSIONS.

Having determined upon the nominal horsepower of the engine, and the fuel to be employed, it is at once approximately known how much of the fuel is necessary in a given time. This determines the amount of air required for combustion under assumed conditions, and in this manner the cubic contents of the charge in any given time, say one minute, becomes known. The generation of a certain horsepower therefore

narrows itself down to the determination of dimensions and speeds which will generate a certain cubic content in unit time. The variables involved under the control of the designer are the diameter of cylinder and the piston speed. The latter in turn resolves itself into stroke and revolutions per minute.

The above method of attacking the problem is much better than to lay out an ideal card, and by the use of card factors to determine the dimensions, as is done, and properly so, in the case of the steam engine. In the case of the internal-combustion engine just these card factors are extremely uncertain, depending upon composition of charge, perfection of ignition and of combustion, proper mixture of charge, etc. In actual practice it is found that these conditions will give a card factor varying from .4 to .8, and it is quite obvious that to weigh all the circumstances likely to attend each case, and from a multiplicity of allowances to determine a proper card factor, is little better than guessing. On the other hand, the first method requires only the determination of the approximate quantity of fuel and of air required for any determined power. Today the characteristics of the various fuels and of their combustion are so well known that no trouble is experienced, and the determination of engine dimensions from weight or volume of charge required would therefore seem to rest on the most natural basis.

In determining the dimensions and speed for any given charge, however, the designer is at once confronted by contradictory conclusions based upon thermodynamic and upon practical considerations. The former call for short stroke and high rotative speed, for under such conditions the time of combustion and of expansion is short, and the loss of heat to the jacket water will therefore be proportionately less. Practical considerations, however, call for long stroke and small diameter combined with the highest possible rotative speed. The reasons may be briefly cited as follows:

1. Short stroke and high speed excessively shorten the time of all events, which may prevent proper mixture of fuel and air, cause poor ignition, and reduce the capacity of the cylin-

der as a pump, if the 4-cycle be used, thus reducing the capacity of the machine.

2. The high compression pressures at present coming into use seriously reduce the cooling surface of the combustion chamber, since the inlet and outlet valve areas are not correspondingly reduced. A long stroke means a correspondingly long combustion chamber for the same end compression, thus furnishing greater cooling surface, and more room in which to place the valves.

3. The machine parts of an engine must be designed to withstand the maximum total piston pressure, while the capacity of the machine is determined by the total mean pressure. The economy of metal will therefore be the greater the nearer the mean pressure is to the maximum. The ratio $\frac{\text{maximum pressure}}{\text{mean pressure}}$ in the ordinary 4-cycle type is not at all favorable, varying in various cases from $\frac{14-18}{1}$, and for this type especially it becomes all important to generate the necessary volume for the required capacity with the smallest possible piston surface, in order to reduce the maximum total pressure, and thus to keep the machine parts as small as possible. Now the reduced piston surface requires an increase in either the stroke or the rotative speed in order to keep the piston displacement, *i. e.*, the capacity of the machine, the same. It is better to increase the stroke, because a higher rotative speed means a higher velocity of rubbing at the crank pin. This in turn means that, if heating is not to result, the rubbing surface must be increased in order to reduce the unit pressure on the pin. Lengthening the pin, however, leads to a greater diameter from considerations of strength. Higher rotative speeds therefore defeat the original design of keeping down the size of the machine parts.

All of this would lead to the conclusion on practical grounds that, contrary to thermal considerations, the ratio of $\frac{\text{stroke}}{\text{diameter}}$ should be as large as possible. The disadvantage of this is that a greater length of machine is required, but the advan-

tages gained in other places more than balance this, at least for stationary machines.

The following derivation of equations for valves of diameter, stroke and revolutions per minute is due to Güldner,* as is also the appended table. All metric measurements and values have been transformed to English units:

Let N_n = nominal brake horsepower.

n = r.p.m.

D = piston diameter in feet.

S = stroke in feet.

$V_h = .785\ D^2S$ = piston displacement per stroke in cubic feet.

$V = e_v\ V_h$ = actual volume of mixture in cubic feet per suction stroke, barometer 28.95 inches, temperature 59 degrees Fahrenheit.

$e_v = \frac{V}{V_h}$ = volumetric efficiency of suction stroke.

L = the volume of air in cubic feet required for 1 cubic foot of gas fuel, or 1 pound of liquid fuel, under most favorable practical conditions.

L_h = The resulting actual quantity of air in cubic feet for one combustion, at nominal horsepower.

C_s = the quantity of fuel used per hour, for gases in cubic feet, for liquids in pounds, for nominal horsepower.

C = the same per horsepower per hour.

C_h = the same per suction stroke.

H = the lower heating value of the fuel, for gases per 1 cubic foot, for liquids, per pound, in B.T.U.

$e = \frac{N_n \times 33{,}000 \times 60}{778\ C_sH} = \frac{2{,}545\ N_n}{C_sH}$ = thermal efficiency at the brake.

From the above we can derive for 4-cycle engines

$$C_s = \frac{N_n \times 33{,}000 \times 60}{778\ He} = \frac{2{,}545\ N_n}{He}; \quad . \quad . \quad (1)$$

*The article may be found either in die "Zeitschrift des Vereines deutscher Ingenieure," April 26, 1902, or in his book, Entwerfen und Berechnen der Verbrennungsmotoren, pp. 213–215.

$$C_h = \frac{2\,C_s}{60\,n} = \frac{2 \times 2{,}545\,N_n}{60 \cdot n\,H\,e} = \frac{84.8\,N_n}{n\,H\,e}; \quad . \quad . \quad (2)$$

$$L_h = \frac{C_s L}{30 n} = \frac{2{,}545\,N_n L}{30\,n\,H\,e} = \frac{84.8\,N_n L}{n\,H\,e}. \quad . \quad . \quad . \quad (3)$$

For 2-cycle engines equations (2) and (3) should be divided by 2, because there is a charging stroke for every revolution.

ENGINES FOR GAS FUEL.

The charge taken in by the engine during one suction stroke will be

$$V = C_h + L_h.$$

This volume requires an effective piston displacement.

$$V_h = .785\,D^2 S = \frac{C_h + L_h}{e_v} = \frac{84.8\,N_n + 84.8\,N_n L}{n\,H\,e\,e_v}$$

$$= \frac{84.8\,N_n (1 + L)}{n\,H\,e\,e_v} \text{ cubic feet.} \quad . \quad (4)$$

Solving (4) for D, S and n in turn, we have

$$D = \sqrt{\frac{108\,N_n (1 + L)}{e\,n\,H\,S\,e_v}} \text{ feet;} \quad . \quad . \quad . \quad . \quad (5)$$

$$S = \frac{108\,N_n (1 + L)}{e\,n\,H\,D^2\,e_v} \text{ feet;} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$n = \frac{108\,N_n (1 + L)}{e\,S\,H\,D^2\,e_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

ENGINES FOR LIQUID FUEL.

In this case the fuel is introduced either in liquid form or in the shape of vapor. In either case the volume ratios of fuel to air are very much smaller than they are in the case of gaseous fuels. Take for instance the case of alcohol, a fuel of low heating value. Here we find that the vapor does not form theoretically more than 4 per cent. of the volume of the

charge. In reality it is even less than this on account of the excess air used. In view of this fact we may put for these engines

$$V_h = .785\, D^2 S = \frac{L_h}{e_v} = \frac{84.8\, N_n L}{n\, H\, e\, e_v} \text{ cubic feet.} \quad . \quad (8)$$

Solving this again for D, S and n we obtain for engine using liquid fuel

$$D = \sqrt{\frac{108\, N_n L}{e\, n\, H\, S\, e_v}}, \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$S = \frac{108\, N_n L}{e\, n\, H\, D^2 e_v}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

and

$$n = \frac{108\, N_n L}{e\, S\, H\, D^2 e_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

In these equations the quantities to which certain values must be assigned are L, e and e_v. As stated before, there is now so much practical data at hand that the proper determination of these quantities should cause no difficulty. Table II* gives various values for e_v, the volumetric pump efficiency, and Table III* values of L as found for various fuels.

Since the nominal brake horsepower N_n is 18 to 20 per cent. below the maximum capacity of engines in ordinary cases, it is well to assume an excess of air of about 30 per cent. at the outset, in the case of machines using gas. In the case of liquid-fuel engines a greater excess, 50 to 60 per cent., should be used, because these fuels are usually high in heating value, and if any economical degree of compression is to be employed, the fuel mixture should be rather lean. Besides, it is not easy to produce a uniform fuel mixture in the case of liquid fuels, and an excess of air would help to overcome this difficulty. These excess air allowances have been made in the table. Table III also gives values of C and e for the various fuels and for various sizes of engines, as determined from existing data. The fuel consumed by heating apparatus

*Güldner, p. 175 and pp. 214 and 215.

and ignition flames is not included in *C*. In the case of engines using generator gas, however, the fuel used by the boiler furnishing steam for the generator has been taken into account, since a part of the heat is recovered from the steam in the generator.

TABLE II.

Type of engine.	Probable suction pressure, pounds.	Values of e_v.
Slow speed. Inlet valve mechanically operated	12.9—13.7	.88—.93
Slow speed. Automatic inlet valve	12.5—13.2	.80—.87
High speed. Inlet valve mechanically operated	11.7—12.5	.78—.85
High speed. Automatic inlet valve	11.4—12.2	.65—.75
Very high speed automobile engines, with automatic inlet valve, and air cooling	8.8—11.0	.50—.65

The ordinary vaporizer for liquid fuels will decrease the values of e_v from 3 to 5 per cent.

It will be of interest to see how the results obtained by use of these empirical formulae and tables agree with the dimensions of existing engines. For this purpose we will take some of the engines in the Mechanical Laboratory and a 140 H.P. producer gas engine tested by Mr. Buckingham in 1903. Two of the variables composing the necessary charge volume, *i. e.*, the stroke and the revolutions per minute, will be assumed, and the third, the diameter, is computed. It will be noted that the agreement is in all cases sufficiently close. It must be remembered, however, that as soon as the engine to be designed differs in any detail to any great extent, some of the assumptions must be modified to suit the altered conditions, and in this is where the experience of the designer finds its application.

TABLE III.

......	Lower heating value (H) B.T.U. per		Air required, cubic feet. Theoretically. (L_0) per		Actually (L) per		Consumption of fuel (C) per brake horsepower hour and thermal brake efficiency (e), when N_n = 5 B.H.P. (C)		(e)	10 B.H.P. (C)		(e)	25 B.H.P. (C)		(e)	50 B.H.P. (C)		(e)	100 B.H.P. & over. (C)		(e)
	cu. ft.	lbs.	cu. ft.	lbs.	cu. ft.	lbs.	cu. ft.	lbs.		cu. ft.	lbs.		cu. ft.	lbs.		cu. ft.	lbs.		cu. ft.	lbs.	
I. Illuminating gas.																					
a. Lean	505	...	5.5	...	7.5	...	24.8	...	.20	22.2	...	.22	20.4	...	.24	19.1	...	.26	18.5	...	.27
b. Ordinary {	560	...	to	...	to	..	22.2	...	.20	20.1	...	.22	18.3	...	.24	16.9	...	.26	16.5	...	.27
	618	...		...		...	20.4	...	.20	18.3	...	.22	16.9	...	.24	15.5	...	.26	15.1	...	.27
c. Rich	675	...	6.5	...	10.0	...	18.7	...	.20	16.7	...	.22	15.5	...	.24	14.1	...	.26	13.7	...	.27
II. Generator gas.																					
a. Basis of anthracite	...	13,400	...	...	...	...	...	1.65	.11	...	1.43	.13	...	1.25	.15	...	1.10	.17	...	1.03	.18
b. Anthracite gas	141	...	.85	...	1.1	...	106.0	...	.17	95.3	...	.19	84.8	...	.21	77.6	...	.23	73.2	...	.24
c. On basis of coke	...	12,950	to	...	to	...	...	1.76	.11	...	1.49	.13	...	1.29	.15	...	1.14	.17	...	1.07	.18
d. Coke gas	129	...	1.00	...	1.4	...	116.0	...	.17	102.0	...	.19	91.8	...	.21	84.8	...	.23	81.3	...	.24
III. Blast-furnace gas	106	...	.75	...	1.0-1.2	...	..	...	...	130.0	...	.18	116.0	...	.20	102.0	...	.22	98.8	...	.24
IV. Coke-oven gas.	505	...	5.3	...	7.0	...	...	...	...	35.3	..	.17	30.0	...	.19	26.4	...	.21	24.7	...	.23
V. Kerosene	..	18,900	...	185.0	...	257–353	...	1.25	.11	...	1.10	.12	...	.99	.13	...	...	...	...	...	...
VI. Crude oil (Diesel)	...	18,000	...	176	...	288–323	...	.55	.25	...	.527	.26	...	.506	.27	...	.462	.30	...	.440	.315
VII. Gasoline	...	19,800	...	176	...	240-323	...	.66	.19	...	.615	.21	...	.550	.23	...	...	...	...	...	...
VIII. Alcohol (90 per ct. vol).	...	10,300	...	96.5	...	128–193	...	1.10	.22	...	.99	.24	...	.92	.26	...	...	...	...	...	...

I. Hornsby–Ackroyd Kerosene Engine.

Brake horsepower = 3.
R.p.m. = 230, normal.
Stroke = 1 foot.

Assume $L = 300$ cubic feet, $e = 8$ per cent., as determined from tests, $H = 18{,}900$ B.T.U. per pound, and $e_v = .80$. The latter is placed at this low figure because the air enters through the heated exhaust ports.

$$D = \sqrt{\frac{108\ N_n\ L}{enHSe_v}} = \sqrt{\frac{108 \times 3 \times 300}{.08 \times 230 \times 18{,}900 \times .80}}$$

$$= .55 \text{ feet} = 6.60 \text{ inches.}$$

Actual diameter = 7 inches.

II. Westinghouse Gas Engine.

2-Cylinder vertical.
Brake horsepower = 4 for each cylinder.
R.p.m. = 300.
Stroke = .66 feet.

Assume $L = 10$ cubic feet, $e = 22$ per cent., $H = 560$ B.T.U. per cubic foot and $e_v = .85$.

$$D = \sqrt{\frac{108 \times 4\ (1 + 10)}{.22 \times 300 \times 560 \times .66 \times .85}} = .471 \text{ feet}$$

$$= .471 \text{ ft.} = 5.65 \text{ inches.}$$

Actual diameter = 5.75 inches.

III. Acme Gas Engine.

Brake horsepower = 14.
R.p.m. = 220.
Stroke = 1.33 feet.

Values of L, e, H and e_v may be assumed, the same as for previous case.

$$D = \sqrt{\frac{108 \times 14\ (1 + 10)}{.22 \times 220 \times 560 \times 1.33 \times .85}} = .76 \text{ feet}$$

$$= 9.12 \text{ inches.}$$

Actual diameter = 9.00 inches.

IV. 140-H.P. Otto Gas Engine.

Brake horsepower = 140.
R.p.m. = 190.
Stroke = 3 feet.

Fuel is producer gas, 135 B.T.U. per cubic foot. Assume $L = 1.3$ cubic feet, $e = 23$ per cent. and $e_v = .85$ per cent.

$$D = \sqrt{\frac{108 \times 140\,(1 + 1.3)}{.23 \times 190 \times 135 \times 3 \times .85}} = 1.52 \text{ feet}$$

$$= 18.24 \text{ inches.}$$

Actual diameter = 19 inches.

IV. METHODS OF REGULATION.

The methods of speed regulation employed in internal-combustion engines may, for the purpose of discussion, be conveniently grouped under the following heads:

1. The hit-and-miss method.
2. Varying the composition of the charge to suit the load.
3. Changing the amount of charge to suit the load.

Other means are sometimes employed. In automobile machines especially speed is regulated sometimes by time of ignition of the charge, the ignition in some cases being completely suppressed. While this method of hand regulation is satisfactory for automobile and marine machines, it can not be said that the attempts to automatically control the time of ignition have been at all successful.

Two points should be kept in mind comparing the above methods of governing. The first of these is their economic efficiency, and the second the efficiency of regulation. Depending upon the service to which the engine is to be put, either one of these conditions may be of the higher importance, for it will in general be found that no one method combines highest economic efficiency with highest efficiency of regulation throughout the series of loads from no load to the rated.

1. The hit-and-miss method is mechanically one of the simplest. Usually a simple pendulum governor controls the

fuel valve, which is either opened or which stays closed during a revolution, depending upon the action of the governor. In some types the mixture-inlet valve is so controlled, and in others the exhaust valve is kept open during the "misses," meanwhile keeping fuel or mixture-inlet valve closed. Theoretically there should be at full load an explosion every 2 revolutions in the 4-cycle engine with this system of governing. At $\frac{3}{4}$ load the order should be: 3 explosions followed by 1 miss, at $\frac{1}{2}$ load 1 explosion followed by 1 miss, and so for all other loads in a regular manner. In general, if x be the fraction of the full load, y be the governing period, *i. e.*, the sum of hits and misses which follow each other other in rotation, and if z be the number of hits in this period, then we may put

$$x = \frac{z}{y}.$$

As long as the resistance encountered at the crank shaft is absolutely constant this law will be followed. But this is never the case, and as soon as the slightest disturbance occurs the governing period and the number of hits will be changed. At $\frac{3}{4}$ load for instance the sequence should be 111—111—111—111—, where 111 represents the hits and — the misses. Any disturbance of the resistance above or below the average may result in the following arrangement 111—11—1111—11—1111—. It can be shown that this at once causes a disturbance in the regulation.* It is of course impossible to control the accidental variations in the resistance to be overcome, and consequently this method of governing will always give somewhat unsatisfactory regulation. For ordinary power purposes this is, perhaps, not of great importance, especially when the prime mover is connected to the shafting or machinery by belting and when the driven machines themselves have masses of rotating metal. In such cases the elastic belts and the machines themselves tend to equalize the speed variations. For other work, however, as for electric lighting, this defect of the

*See Güldner, Verbrennungsmotoren, p. 317, and Mollier, "Zeitschrift des Vereines deutscher Ingenieure," p. 1704, Nov. 21, 1903.

hit-and-miss system is much more serious. It is of prime importance in such cases to carefully construct not only the governor, but also the mixing valves and the igniter, in order to make the number of misses between any two explosions as small as possible, and to be sure that the explosion following a miss occurs at the proper time with full force. But even under such conditions, it is not likely that any thing like satisfactory regulation for electric service, etc., can in a 4-cycle machine be attained by the hit-and-miss method, and Güldner advises to make the weight of the fly wheel rim twice as heavy as computations with allowable variation at full load require. This can, of course, not be commended from the standpoint of economy of first cost, and we find therefore that this method of governing is confined mostly to engines where the regulation need not be closer than 2½ per cent. from full load to no load.

On the other hand, this method of governing is more economical of fuel than any other yet devised. The reason for this is apparent. If at full load the conditions, *i. e.*, amount of compression, composition of charge and point of ignition, are such as to give the highest efficiency for the fuel used, this efficiency should remain constant for each cycle from full load to no load, since none of the conditions, under which each cycle is described, are changed. Conditions extraneous to the working of the cycle of course change this, and we find that thermal efficiency decreases with the load. The fact remains, however, that, unless the conditions are abnormal, the efficiency is better sustained throughout the range of load in this system of governing than in any other. This is undoubtedly true for engines using gas or gasolene. In kerosene and alcohol engines it is quite probable that some of the fuel vapor is condensed on the comparatively cool cylinder surfaces after a series of misses at low loads, causing an increase in the fuel consumption, and a consequent decrease in efficiency.

2. Governing by varying the composition of the charge to suit the load is a method often employed, and, with proper fly-wheel weight, is found to give good satisfaction as regards

regulation. From the standpoint of fuel economy it is somewhat inferior to the hit-and-miss method. The same charge volume is used every cycle, the composition being changed by governor control of the fuel valve, the richer mixtures being used for the higher, and the leaner ones for the lower loads. The charge volume remaining constant, the amount of compression remains the same throughout, and the same thermal efficiency should therefore be realized from all cycles. As the load decreases, however, ignition of the leaner mixtures becomes increasingly difficult, and combustion is less complete and rapid. At the lower loads, cards from engines governing in this way sometimes show expansion lines which sink very slowly, showing an unusual amount of after burning.* Attempts have been made to avoid this by a simultaneous adjustment of the point of ignition, making it earlier, but to the writer's knowledge, this has not so far been very successful. We find, therefore, rather high fuel consumption at the lower loads in engines governing by this method, and the tendency today seems to be to desert this for other methods.

3. The third one of the principal methods of governing is to change the amount of the charge to suit the load, leaving its composition practically the same. This may be done in two ways. The mixture may be throttled throughout the suction stroke, in which case the pressure in the cylinder constantly decreases from beginning to end of suction. At full load with little or no throttling we shall have the regular card, as indicated in Fig. 1, for the 4-cycle machine. As the load decreases, the amount of throttling increases until we have a card as indicated in Fig. 2. The second way is to cut off the mixture completely some time before the suction stroke is complete, the time of cutting off depending on the load. In this case, too, we shall at full load have a card like Fig. 1, but at some lower load the card will be like Fig. 3, cut off occurring at b'. The second method is better than the first because less work is lost in the lower loop. They both possess the disadvantage that the compression varies with the

*Meyer, "Zeitschrift des Vereines deutscher Ingenieure," p. 602–603, April 25, 1903.

load, decreasing as the load decreases. Consequently, everything else being the same, as the load decreases the engine works with cycles whose efficiency constantly decreases. This defect in this method of governing is to some extent compensated for by the fact that more complete expansion is obtained as the load decreases, and it may be said that, considering both efficiency of regulation and economy of fuel, this method is the best of the three. For closeness of regulation this is undoubtedly true. The question as to how much inferior this method is to the hit-and-miss system as regards fuel economy led Meyer to make the following interesting comparison between the two, in connection with his Report on Tests of Alcohol-locomobiles for the German Agricultural Society, 1902.* Tests were made on a Deutz engine using the cut-off system of regulation, and the following figures, transposed to English units where necessary, were obtained.

Load.	R.p.m.	Brake horsepower. (Metric.)	Per cent. of full load.	Vacuum at end of suction stroke.	Pressure at end of compression above atmosphere.	Per cent. cut-off = $\frac{a'd'}{a'i'}$ Fig. 3.	Mean effective pressure of the area $a'b'c'd'i'h'a'$ Fig. 3.	Work lost in per cent. of net area of diagram.
				Pounds.	*Pounds.*		*Pounds.*	
Full	278	15.95	100	2.40	201.0	93	2.70	.10
¾	284	12.18	74.5	5.25	134.5	71	2.55	1.80
	286	10.26	62.4	5.80	124.5	66	2.55	2.90
½	286.5	8.18	49.8	6.80	106.0	56	2.70	5.40
	289.5	6.19	37.3	9.06	88.0	42	2.55	12.10
¼	294	4.22	25.0	10.05	56.8	33	2.42	20.30
	296	2.07	12.2	10.75	54.0	24	2.13	34.3
Friction	....	0	0	11.50	31.2	19	1.84	48.4

Meyer makes the comparison by determining how much work is lost when a given amount of mixture is in the one case worked through cycle Fig. 1, *i. e.*, the hit-and-miss system, and in the second case through cycle Fig. 3, *i. e.*, the cut-off system. The areas of the cards measure the work de-

*Meyer, "Zeitschrift des Vereines deutscher Ingenieure," p. 602–603, April 25, 1903.

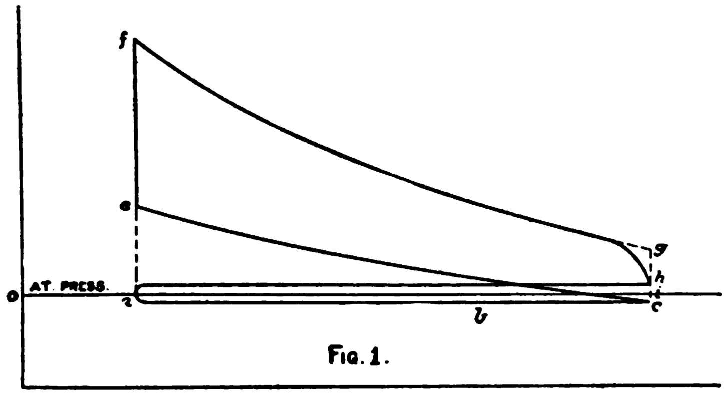


FIG. 1.

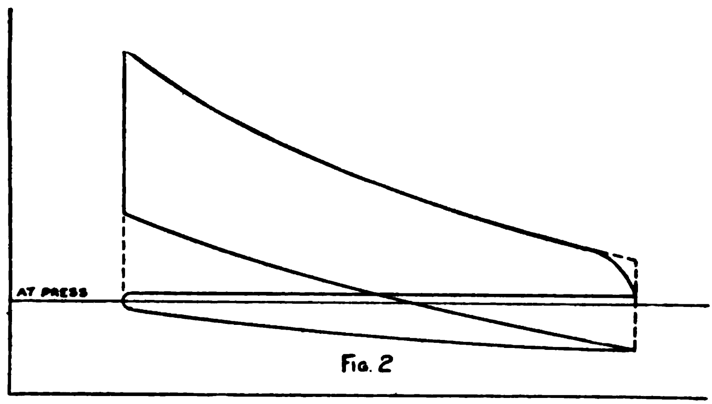


FIG. 2

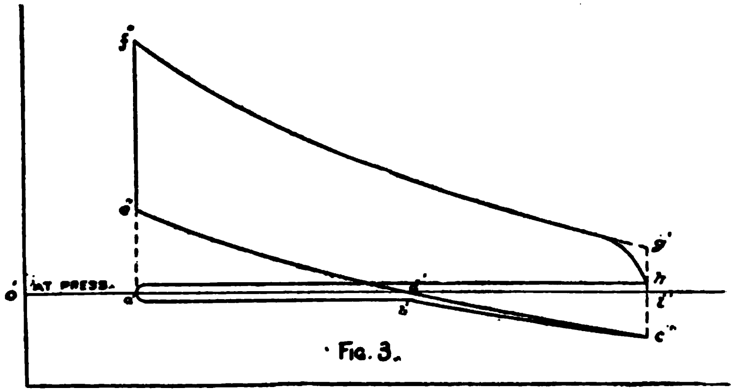


FIG. 3.

veloped, so that the comparison may be made on the basis of areas. In each cycle, part of the work is lost in exhausting the burned gases and in taking in a new charge. The work thus lost may be measured by the mean effective pressure of the exhaust and charging strokes. It is in Fig. 1 indicated by area *a b c h a* and in Fig. 3 by the area $a'\ b'\ c'\ d'\ i'\ h'\ a'$. On the tests of the Deutz engine, above mentioned, Prof. Meyer took a weak spring card at a number of loads, and by measurement found the mean effective pressure of the various areas $a'\ b'\ c'\ d'\ i'\ h'\ a'$. These are given in column 8 of the table. It will be noted that down to less than ¼ load the mean effective pressures are nearly constant, *i. e.*, the work lost is practically constant. But at full load, in Fig. 3, the loss is the same as in Fig. 1 at any load, and finally, therefore, the exhaust and suction loss is practically the same for the two systems. The next area to be considered is that generated by the cycle proper. It is assumed that the specific heat of the gases is constant, and that the cycle is reversible. Further, that the same amount of mixture, *i. e.*, heat, is added along *e f* and $e'\ f'$, and that $c'\ e'$, $f'\ g'$, *c e*, *f g* are adiabatics. Under such conditions, if the ratio $\frac{o\ c}{o\ a} = \frac{o'\ c'}{o'\ a'}$, the efficiencies of the two cycles will be the same irrespective of the position of the point c' with reference to the atmospheric pressure line. This means that area $c'\ e'\ f'\ g'\ c'$ = area *c e f g c*. In reality, the specific heat of the gas not being constant and the cylinder walls conducting heat, area $c'\ e'\ f'\ g'\ c'$ will be somewhat greater because the average working temperature is less in Fig. 3 than in Fig. 1. Neglecting this, however, the *net* area generated will be the same in the two cases. But of the area $c'\ e'\ f'\ g'\ c'$ the part $c'\ d'\ i'\ c'$ is not utilized in the cut-off system, and this would then seem to be the only net loss in this system over the hit-and-miss method. Expressing this loss in per cent. of the net area, we shall have

$$\text{Loss} = \frac{100 \times \text{area } c'\ d'\ i'\ c'}{\text{area } c'\ e'\ f'\ g'\ c' - \text{area } a'\ b'\ c'\ d'\ i'\ h'\ a'} \text{ per cent.}$$

This quantity, column 9 of the table, is also a measure of the heat lost in the cut-off system of regulation as compared with the hit-and-miss system.

In commenting upon the figures, Prof. Meyer remarks that down to ½ load the economy of the cut-off regulation may be considered practically equal to that of the hit-and-miss regulation. Below this the economy rapidly falls off, until at friction load the fuel consumption would be about twice that of the hit-and-miss regulation. He points out the fact, however, that at low loads in the hit-and-miss system, conditions extraneous to the cycle enter, which tend to increase the fuel consumption. The main one of these is probably, in oil and alcohol engines, the cooling of the cylinder walls during a series of misses leading to the condensation of fuel vapor on the next explosion. So that in practice the results are probably somewhat more favorable at loads less than ½ than the table indicates. The general conclusion of the comparison would be that on the score of speed regulation the cut-off method is the best, and that with reference to fuel economy this method is nearly on a par with the hit-and-miss system in ordinary commercial operation, it being assumed that operation below ½ load for any length of time is not an ordinary condition.

While it is thus shown that the cut-off method is but little inferior, attempts have been made to improve it by changing the clearance space with a variation in the cut-off, in order to keep the compression the same at all loads. No satisfactory motor of this type has so far, however, been designed.

Among the various modifications of the systems of speed regulation above described, that used by Letombe deserves special mention. In this method both the composition of the mixture and the charge volume are changed. The governor operates to make the mixture leaner and the cut-off later as the load decreases. In this latter respect it works opposite from the ordinary way in which the cut-off grows earlier as the load decreases. The result of this regulation is that the leaner gas mixtures are the more highly compressed, which

is thermodynamically desirable. The lean composition of the mixture makes any danger from pre-ignition remote. Being a thermodynamically more correct system of governing, better results might have been expected, but tests have shown that while the results attained are very good, they are no better than those attained by first-class engines using other systems of speed regulation. (See Güldner, Verbrennungsmotoren, p. 424.)

V. ALCOHOL AS A FUEL.

For the American power user the question of the utilization of alcohol in internal-combustion engines has at present very little commercial interest, and this state of things is likely to continue for a long time. For many European countries, more particularly Germany, the conditions are, however, very different. There crude oil, kerosene and benzine, owing to duty, are nearly on a level in price with alcohol, which is there a very important domestic product. As will be seen below, alcohol can be bought in moderately large quantities even cheaper than the other fuels mentioned. Germany also exhibits the rather queer spectacle that the engine builder did not come to use alcohol because he recognized in it a good fuel for his engine, but because the distiller, owing to overproduction, had to find use for his product. This condition of things is not natural, and there is, consequently, some question as to its stability.*

However that may be, and whatever the present-day interest of the American power consumer, the manner in which the problem of the utilization of alcohol was solved, the results attained and the present status of the question should be of interest to the American engineer.

Since 1887 several things concurred to cause an annual overproduction of alcohol in Germany. The potato crop, from which in the Eastern provinces most of the alcohol is made by agricultural proprietors, increased about 33 per cent. during the last ten years. An increase in the revenue tax on

* R. Schöttler, "Zeitschrift des Vereines deutscher Ingenieure," p. 1157, Aug. 2, 1902.

alcoholic beverages lowered the domestic consumption of alcohol, and an increase of import duty imposed by foreign countries caused a decrease in the exports. The potato crop, after the demand for them as food has been satisfied, is of little value except for the making of alcohol; under these circumstances the supply rose while the demand decreased. The natural outcome was to seek new fields for the utilization of the product, and in consequence the amount of alcohol used for technical purposes increased from about 10,000,000 gallons in 1887 to 30,000,000 gallons in 1901.

The price naturally decreased during this period, but it fluctuated so much that no decided step formed in the utilization of the product as fuel for gas engines was made until about 1899. Today, under a schedule of prices fixed by what appears to be a company controlling the alcohol supply, 90 per cent. alcohol can be bought in the consumer's own vessels, and in lots of not less than 1,600 gallons, for $3.51 per 26.4 gallons, which is about 13.3 cents per gallon. This price prevails from November 1 to May 15; for the remainder of the year the price is $3.74 for the same quantities, corresponding to 14.2 cents per gallon. It is guaranteed that these prices will be maintained until 1908. Under such conditions alcohol can compete with benzine, gasolene or kerosene, at the prices prevailing in Germany for the latter fuels. As Schöttler remarked, however, and as some engineers of authority, Diesel, for instance, strongly believe, the condition of things is unstable, since nobody knows what will occur after 1908. Diesel* seems to believe that alcohol will never be a serious competitor of either benzine, gasolene, kerosene, or crude oil, even in Germany.

In comparing the various liquid fuels above mentioned, regard should be paid to the following points:

1. Cost of the various fuels on the basis of heat units, say 10,000 B.T.U.

2. Cost per delivered horsepower per hour.

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*" Z. d. V. deutscher Ingenieure," p. 1367, Sept. 19. 1903.

1. COST OF THE VARIOUS FUELS ON THE BASIS OF HEAT UNITS.

Favre and Silberman determined the lower heating value of absolute alcohol to be 11,664 B.T.U. per pound. Commercial alcohol is, however, never absolute, containing various quantities of water. Besides this, the German law requires that alcohol used for technical purposes in order to be free from revenue tax shall be rendered undrinkable by the addition of about 2 per cent. wood alcohol and ½ per cent. of Pyridine bases. These admixtures change the heating value. Taking into account the water only, it lowers the heating value in two ways, first by replacing some alcohol, and secondly, by requiring heat for its own vaporization. The following table, due to Schöttler, gives the variation in the heating value and in the cost of alcohol of varying purity. The price is taken at about $3.74 per 26.4 gallons for the 90 per cent. alcohol.

Absolute alcohol, volume, per cent.	Spec. gravity.	Absolute alcohol weight, per cent.	Lower heating value. B.T.U. per pound.	Cost, cents per gallon.	Cost, cents per pound.	Cost of 10,000 B.T.U., cents.
95	.805	93.8	10,880	16.3	2.44	2.24
90	.815	87.7	10,080	15.4	2.28	2.25
85	.826	81.8	9,360	14.5	2.12	2.26
80	.836	76.1	8,630	13.6	1.96	2.27
75	.846	70.5	7,920	12.8	1.82	2.29
70	.856	65.0	7,200	12.0	1.69	2.35.

Comparing the above figures with the cost of gasolene and kerosene, Meyer gives the price of gasolene as 24 Pf. per kg., which is 2.64 cents per pound, or about 15.5 cents per gallon, the cost of kerosene is given as 22 Pfg. per kg., which reduces to 2.43 cents per pound, or about 16.2 cents per gallon. The heating value of these two fuels is about the same, and may be taken at about 18,500 B.T.U. per pound on the average. This will make the cost of 10,000 B.T.U. for gasolene 1.43 cents, and for kerosene 1.31 cents. Diesel gives the price of paraffin oil, or crude oil, for Germany at from 8.25 to 10 Pfg. per kg., which corresponds to from .88 to 1.06 cents per

pound. He refers here to domestic oils, the cost of the imported product being higher on account of import duty. Assuming 18,000 B.T.U. as the average heating value of the crude oil, the cost of 10,000 B.T.U. will be from .48 to .60 cents.

The following table gives a summary of all the figures. For the purpose of comparison the cost figures for the U. S. have been appended. For these columns, the prices were taken as follows :

Gasolene 13.5 cents per gallon in barrels of from 50 to 56 gallons.

Kerosene 12.5 cents per gallon in barrels of from 50 to 56 gallons.

Crude oil, $2.50 per barrel of 50 gallons at New York.

The figures for alcohol are not given, for with ethyl alcohol at $2.50 per gallon, and methyl alcohol at $1.25, there can be no rational basis for comparison.

......	Cost per gallon, cents.		Cost per pound, cents.		Cost of 10,000 B.T.U., cents.	
	Germany.	U. S.	Germany.	U. S.	Germany.	U. S.
Gasolene....................	15.5	13.5	2.64	2.32	1.43	1.25
Kerosene....................	16.2	12.5	2.43	1.88	1.31	1.02
Crude oil, sp. gr. = .9..	8.0	5.0	1.06	.66	.60	.36
Alcohol, 90 per c. vol...	14.2	...	2.28	...	2.25	...

2. COST PER DELIVERED HORSEPOWER PER HOUR.

The highest economy so far shown for alcohol was obtained on the tests of the Deutz alcohol motor by Prof. E. Meyer. For purposes of comparison he added to these figures the highest economy obtained with gasolene and kerosene engines. Diesel quotes this table in an article in the "Zeitschrift des Vereins deutscher Ingenieure," Sept. 19, 1903, and adds to it the best economy figures obtained by the Diesel engine on crude oil. The annexed table shows the results transfer-

red to English units. Diesel remarks that on later tests the consumption of his engines has been decreased about 15 per cent. under that given in the table.

		Other cycle engines.			Diesel Engine.
		Gasolene.	Kerosene.	Alcohol.	Paraffin oil.
Cost of fuel per lb., cents		2.64	2.43	2.28	1.06
Cost of 10,000 B.T.U., cents.		1.43	1.31	2.25	.60
Ratio, crude oil—1		2.40	2.20	3.75	1.
Best Consumption lbs. per H.P. hour	Full L'd..	.654 (Banki, .488)	.727	.805	.450
	Half L'd..	.955	1.082	1.117	.533
Increase in consumption, full load to half load, p. c..		46.	49.	44.	19.
Cost per Delivered Horsepower per hour, cents	Full L'd..	1.73	1.77	1.83	.477
	Half L'd	2.52	2.64	2.54	.565
	Average..	2.12	2.20	2.19	.521
Ratio, crude oil = 1		4.06	4.21	4.20	1.
Hig'st thermal efficiency per D.H.P. so far obtained	Full L'd, p.c	20.5	17.6	32.7	32.6
	H'f L'd, p.c	14.0	11.8	22.7	27.4

The table shows in the first place that crude oil, as burned in the Diesel engine, is the cheapest of the fuels, the ratio being about 1 to 4 in its favor. This superiority probably holds good also when this oil is burned in an explosion engine, in contradistinction to the Diesel, provided that such an engine can successfully handle this fuel. In a test made on an engine using the Otto cycle by the writer in New York during the summer of 1903, for instance, the consumption of oil for full load was .488 pounds per B.H.P. per hour, and for half load, about .735 pounds. This, at the rate of 5 cents per gallon for the oil at New York, corresponds to a cost of .321 cents per B.H.P. at full load, and .485 cents at half load, an average of .403 cents. Crude oil would therefore seem to be the fuel of greatest promise, but there are a great many practical difficulties in the way of its successful application, although it is claimed that it is burned successfully in a number of types of engines.

The table also shows that in spite of the fact that 90 per cent. alcohol costs about 60 per cent. more per 10,000 B.T.U.

than either gasolene or kerosene, as regards the cost per delivered horsepower, *when each is burned in an engine designed for its use*, they are all on the same level. The reason for this is not far to seek. The thermal efficiency of the Deutz engine was 32.7 per cent., as compared with 20.5 per cent. for the gasolene and 17.6 per cent. for the kerosene engine, that is, a properly designed alcohol engine will utilize 50 per cent. more of the heat furnished it than a gasoline engine, and nearly 100 per cent. more than this last engine when using kerosene. In explaining this result, Meyer determined first that the work lost in fluid friction, friction of the machine, etc., was about the same in the alcohol engine as it was in the one which was used for the gasolene or kerosene tests. He further showed, that since the degree of compression was the same for both the kerosene and gasolene tests, the greater consumption of the kerosene can only be explained on the assumption that some of the kerosene goes through the engine unburned, the vapor being condensed as it enters the cylinder. This being so, the greater thermal efficiency of the alcohol engine can only be due to differences in the cycle, and here we find that a much higher degree of compression can be employed with alcohol than with gasolene or kerosene. This is due to the fact that the mixture of alcohol and air possesses a rather high specific heat, the percentage of water contained in the alcohol further increases this, and in consequence a greater degree of compression can be used without the fear of premature explosions. While the ratio of compression for gasolene or kerosene can not be carried much above 4, that for alcohol may with ease be made 10, and this in itself would mean on theoretical grounds an increase in the efficiency equal to about 50 per cent. As is well known, Banki increases the ratio of compression for gasolene by injecting water vapor, and has in that way reached a degree of compression equal to 9.83, the consumption of gasolene being .488 pounds per B.H.P. per hour, corresponding to a cost of 1.29 cents per B.H.P. per hour. Against such results alcohol will, of course, never be able to compete, but aside from this it is in Germany practically the equal of gasolene and kerosene as regards economy.

For automobile purposes, however, the Banki principle can not be used on account of the large quantities of water which must be carried, and here, as also for general power purposes, alcohol possesses several advantages over the other fuels. It is practically odorless, and much less inflammable than gasolene. As compared with korosene, it is much cleaner, there being no carbon deposit formed. Air even at ordinary temperatures takes up considerable alcohol vapor, so that the danger of condensation in the cylinder is minimized. There is, of course, danger of rusting through the use of alcohol, but proper care should render this disadvantage of little moment. Rusting may also be prevented by the addition of benzol (C_6H_6), and tests have shown that the cost of operation with a mixture of 15 per cent. by weight of benzol and 85 per cent. of alcohol is a little lower than the cost when alcohol alone is used. It has, however, been noticed that engine needs cleaning more often when such a mixture is used. A further disadvantage in the use of alcohol lies in the fact that it must be vaporized through heat, and it therefore becomes necessary either to have an open flame to preheat the vaporizer, or to start with gasolene. The latter method is preferred to the former, there being less danger, in the case of traction engines or other agricultural machinery, and also because the change from gasolene to alcohol can ordinarily be made by throwing over a single lever.

While Meyer's results thus point to the practical equality of alcohol as a fuel for gas engines, he himself has pointed out that under the high degrees of compression employed, the engines are very sensitive as regards the proper amount of fuel taken per stroke, the explosions easily becoming excessive, and he asks whether it would not be better to use somewhat lower compression to insure greater certainty of operation, of course at the cost of economy. As a matter of fact, comparative tests made in Nürnberg* on a 2-H.P. gasolene engine in which the compression could be changed from 50 pounds to somewhat

*" Z. d. v. deutscher Ingenieure," p. 59, Jan. 10, 1903.

above 105 pounds have led to different results. The conclusions from these tests are briefly as follows:

1. The consumption of alcohol is even at the high compressions greater than the consumption of gasolene.

2. In order to approach the operating cost of a gasolene motor, the alcohol motor requires a much higher compression. This has the disadvantage that a much heavier engine is needed to stand the work.

3. The mixture of alcohol vapor and air is much more difficult to ignite than the gasolene vapor. It is therefore necessary to employ gasolene on starting.

4. The tests have not shown that less fouling of the engine takes place with alcohol than with gasolene. On the contrary, a great deal of water was found in the ignition canal and exhaust passages after long operation with alcohol. Analysis of the exhaust gases also showed small quantities of acids, as acetic acid, whose action on the iron parts can be but unfavorable.

Of these conclusions, the second is well taken, and has its greatest bearing in the case of automobile machines; the third has already been mentioned. The first can perhaps be explained on the ground that the engine was much smaller than that used by Meyer; the fourth, however, is in direct contradiction to the experience of other experimenters. On the basis of these experiments the final conclusion is reached "that equality of operating cost as between alcohol and gasolene is reached only when the compression pressure in the alcohol motor is at least 120 pounds per square inch, and even then 20 per cent. of benzol must be added. The alcohol engine can not seriously compete with the gasolene engine, especially for automobile purposes and other small units."

Diesel also attacks the use of alcohol, and on double ground. Referring to his engine, which he claims can burn any of the liquid fuels to equal advantage, he states that price of alcohol would have to decrease in the ratio of 4 to 1 in order to compete with the Diesel engine. This can not, of course, be expected; but by the same process of reasoning the

same conclusion also applies to gasolene and kerosene. The next statement, that even if Germany produced no paraffin oils, the operation of the Diesel engine with the taxed imported kerosene would cost only half as much as the operation of an alcoholic engine, carries more weight. He states also that the present price of alcohol in Germany is artificial and causes a loss to the producer, the loss being evened up by the higher cost of alcohol not used for technical purposes, *i. e.*, at the cost of third parties, and concludes by pointing out that, in nearly every other country, with a few exceptions, the cost of the liquid fuels is so much less than in Germany, that alcohol would have to cost very much less than it does now in order to enter into competition. Taking crude oil at 5 cents per gallon at New York, for instance, alcohol would have to cost about 2.5 cents per gallon in order to cost the same on the basis of heat units.

PRACTICAL AIDS FOR THE NAVAL ENGINEER.

By Lieutenant H. T. Winston, U. S. N., Member.

All officers who have been assigned to engineering duties in the Navy, whether they be young, and have had little previous engineering experience, or old and experienced engineers, particularly if they have been on shore duty for some time, must realize that there is something lacking in the available text books, articles on engineering, and the engineering instructions of the Navy Regulations. These text books, though excellent in their own line, and which a good engineer must be familiar with in order to understand his machinery and its care, are intended primarily for midshipmen, and they fail to give in concise form a practical scheme for the proper internal workings, the coordination of different subdivisions, and for the day-to-day work of the engineer's division.

It will be the purpose of this article to give such a scheme, and in addition a few sketches and tables, all of which have been tried out by the writer and found to be of much benefit, in so far as smooth working of the engineer's department is concerned. In addition, it has been found that the men become much more proficient, a heavy tax is removed from the engineer's memory and he has considerable more time for study, for other important duties, or for leisure if he so elects.

While in some respects what follows applies only to a large vessel, and particularly to the *Charleston*, with her four-watch system, twin screws, four firerooms and four 4-furnace B. & W. boilers in each fireroom, still it is believed that this article is for the most part of general application, and its purpose will have been accomplished if suggestions of value

have been made to engineer officers, though they may change the scheme to suit their own ideas and to meet varying conditions.

One of the most successful and largest contracting firms in New York, which erects sky-scrapers and contracts for similar work, gives to each employee in its head offices a typewritten copy of the rules which are to guide him in his work. A preface to each set of rules states that their issue has been made necessary by the fact that, whenever anything has gone wrong in the office, an investigation showed some of these rules had not been observed, and that it was desired to avoid in the future the consequent "unpleasantness" for the employee. As in private business, so in the Navy, written instructions serve to avoid "consequent unpleasantnesses," particularly as the men are often green and have had little opportunity to acquire knowledge of routine work, except through long experience and hard knocks.

ENGINE-ROOM AIDS. (See Appendix A.)

The officer of the deck of the engineer's department is usually at the desk in the starboard engine room, and near by should be found the following aids:

Port and Sea Routines (preferably in a bulletin board).

Book of Routine Orders (hanging over desk).

Morning Order Book (contains for the night and following morning orders additional to the routine and routine orders).

Muster Lists of Steaming Sections (in the desk).

Tables of Water in Double Bottoms, per inch, as sounded (in the desk).

Booklet of Skeleton Pipe Plans (in the desk).

Table showing Heaters and Pantries on each Heating Circuit (hanging up).

Diagram of Drainage System and Accompanying Table (in a bulletin board).

A Clinometer so situated that it is easily seen.

On a board (about 18 inches by 12 inches) post auxiliary

watch list, lists of special working parties or details, sick, confined, absentees, cooks, and special notices.

NOTE.—As the notices for this last board are frequently changed, it will be found that a piece of ½-inch oak or pine, studded with nails driven through from the back so the points project about ¼ inch and are 2 inches apart, makes a very convenient kind of board.

DUTIES OF YEOMAN AND SPECIAL DETAILS.

In appendix "B" are given rules for the engineer's office, storerooms, etc. It is suggested that the different sets of rules be posted where the special details and others stand their watches or perform their duties. By doing this there will be a similarity in the performance of duty by the different men, and any new man on a station will have before him general rules for his guidance and will be unable to plead as an excuse ignorance of previous orders.

The "Duties of the Yeoman and his Assistants" is deserving of special attention on the part of the engineer officer, as it contains many things the senior engineer must remember. These items have been purposely omitted from "Aids for the Engineer's State Room" in order to avoid duplication.

Perhaps some officers will think that the keeping of a "Coal-bunker expenditure book" is a useless amount of trouble, but I have used this system on two ships (*Charleston* and *Newport*) and found that it was about the only satisfactory way in which to keep the coal account straight. After a few days the yeoman becomes accustomed to this method, and work is really saved instead of added, because the books show the actual amount on hand. It might be noted further that with a separate coal account for each fireroom, mismanagement or poor firing in that fireroom or an incorrect coal tally shows up at once. As a parting shot, I cannot help adding that tally boards for the number of buckets of coal should, of course, be used in each fireroom.

ENGINEER'S BULLETIN BOARDS.—(OUTSIDE ENGINE ROOM.)

The subject of watch, quarters and station bills will not be taken up in this article, with the exception of a few suggestions, since they must vary considerably on different vessels, and this matter has already been treated by writers in the JOURNAL and by Commander Barton in "Naval Engines and Machinery." The suggestions follow:

Make the bills as simple as possible; let the "steaming sections" be the framework of the whole structure, with the men at their steaming stations for all duties as far as possible; in changing from steaming to auxiliary watches do not change the men on watch, but simply reduce the number (and *vice versa*); have the liberty lists so made up that men will go ashore when off watch duty and will stay on board when on watch duty.

A very convenient form for the bills is one which allows the easy shifting of men's stations. This may be provided for by having each man's name on a separate piece of card board held in place by brass clips so the names may be shifted. By making these clips long enough and having other pieces of card board on which are printed "sick," "absent," "confined," etc., it may be noted at once whether a man is available for duty or not.

It is hardly necessary to add that all bulletin boards should be located where they are conspicuous and may be easily consulted by all members of the engineer's force. Some of them also will require frequent visits of the yoeman, and should be near the engineer's office.

One or two special bulletin boards will be found necessary for the posting of notices, orders affecting the whole division and miscellaneous information; for example, "sea and port routines," "liberty lists," "special requests for liberty to be made before 11·00 A. M.," "steaming shoes to be taken off before going on deck," etc.

AIDS FOR SENIOR ENGINEER'S STATE ROOM.

Appendix "C" contains matter which will be found useful to the senior engineer personally, and may be kept in his room for ready reference.

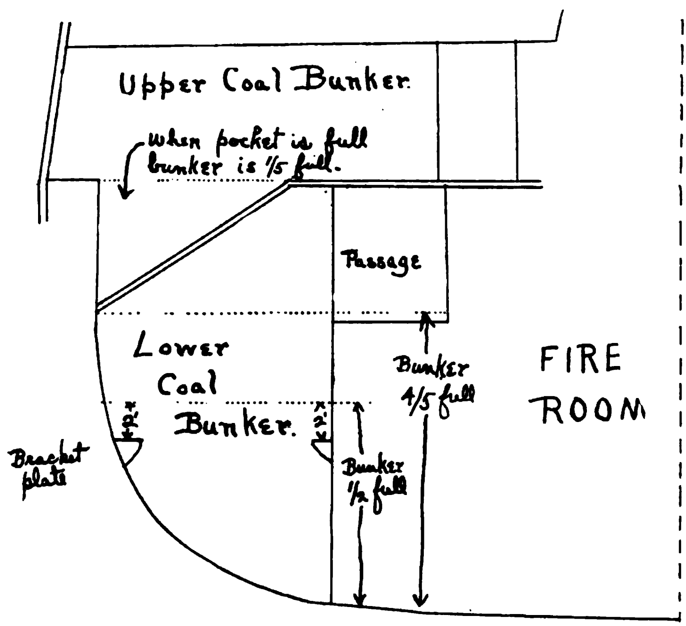
The *routines* show what he must attend to apart from what has been stated heretofore, and may also be used as check lists, the different items being checked off as the work is done and the dates noted if so desired.

The *coal and water consumption* at various revolutions is often a vexatious problem, and the table given will be found useful in solving the problem. (The writer uses curves from which this table has been taken.) With the data from the progressive or measured-mile trials of the ship it is a simple matter to construct the necessary curves and with them to tell approximately how much coal and water will be used at a given speed, except at very high or low speeds. Also taking the amount actually burned in a day it may be seen at a glance whether one has an economical performance or not.

For this ship (*Charleston*) it has been assumed that ordinarily the auxiliaries (apart from the main engine and its necessary auxiliaries) require at sea an average of ten tons of coal per day, which is perhaps a rather low estimate, since the coal consumption in port averages about fifteen tons. The small amount of accurate data available shows that the actual coal consumption at sea runs, under ordinary conditions, slightly above and almost parallel to the 2 pounds per I.H.P. curve. In this connection, it may be interesting to note that the steam consumption of the *Charleston's* port engine was increased about 30 per cent. by a badly scored H.P. cylinder liner and scored liner of H.P. valve chest. A calculation of the steam consumption from indicator cards taken from both engines when running at 90 turns (nearly one-fourth full power) gave the above result, and steam gauges showed that the port H.P. valve leaked so badly that the drop in steam pressure in H.P. cylinder was only about 8 pounds.

"*Books and Notes for Reference*" gives a list of what the engineer will find desirable and useful in looking up various subjects.

Estimating coal in the bunkers is at the best an approximation, and some "landmarks" in the bunkers, particularly if they have an irregular cross-section, will aid in making the estimate fairly accurate. The sketch shows a sample of such "landmarks," the capacities up to them being calculated from measurements taken from the ship's drawings, and assuming that coal can be stowed to within nine inches of the deck overhead.

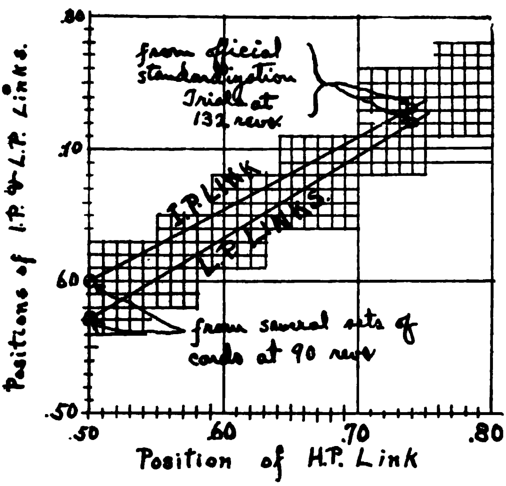


Aid for Estimating Coal in Bunkers.

Fig. 1.

Setting the links to obtain proper distribution of power is another proposition which must be tackled by the engineer To obtain the most economical performance the H.P. links will be run in as far as possible and the throttle kept practically wide open. While the writer has not experimented sufficiently in this line to assert that the curves shown are entirely correct, he has found them of considerable assistance on the

Charleston, where the speed has not varied greatly during a cruise around South America. It is believed that by running with the throttle wide open, adjusting the H.P. links to give two or three different speeds, and setting the other links to give an even distribution of power, fairly accurate curves could be plotted from the points obtained.



Curves for Setting Links so as to obtain Equal Distribution of Power.

Fig. 2.

The straight lines have checked up closely for a few different speeds with the *Charleston*. Of course, it should be remembered that the pressure in the H.P. valve chest must be constant, and that different results will be obtained with steam on and off the jackets. The rule given by Lieutenant H. C. Dinger, U. S. N., in his article in Vol. XVIII, No. 4, p. 144, of the JOURNAL, with regard to estimating the horsepowers of the different cylinders of the engine, holds only rather roughly when there is much drop from the L.P. valve chest to the condenser and from the I.P. to L.P. valve chests. Such is the case on the *Charleston*, and the horsepower of L.P. cylinders is largely overestimated from the gauge readings.

MISCELLANEOUS NOTES.

There remain in addition to the above many other questions which may be perplexing, particularly to the line officer doing engineering duties for the first time. Among these are what duty will be required of certain special mechanics. For example, the coppersmith should understand that he is particularly responsible for all copper piping (especially salt-water lines), that he is to keep properly repaired all oil cans, oil trays, oil lights, oil piping, dust protectors of auxiliary machinery, etc. The blacksmith should keep in a proper state of repair all boiler-firing tools, lathe tools, coal shovels, wrenches, etc.

Throughout the engineer's department numerous things may be done to aid in the smooth and orderly running and the tidy appearance of the department. Some of these are the proper distribution of tools and wrenches which belong to certain stations or machines, as evaporator room, machine shop, steam launches, fireroom, fireroom blowers and ice machines. Each station should have its own tools, properly marked, and the leading man on the station should be responsible for them. Should anyone desire other tools they must be taken from the store rooms and not from other stations. Locked tool chests in each fireroom and on other stations will be found most useful in preventing the loss or mislaying of tools.

Small lockers, even if they are little more than pigeon holes, for steaming shoes in the engine-room hatch are a great convenience for the men and prevent trouble about dirtying up decks.

Good canvas screens for the crankpits, with suitable stops, heavily painted on one side with asphaltum and on the other with brown zinc, tidy up the engine room in addition to keeping the floor plates dry.

Suitable permanent clothes lines, located where the clothes will not be soiled or water drip on machinery, are easily rigged They will be found a convenience to the men, and with them clothing will not be scattered around the Department.

Considerable attention may be paid to auxiliaries both in and outside the Department. Some of them use an excessive

amount of steam even when well cared for (see Commander W. W. White's article on tests on board the *Minneapolis*). With improper care this consumption of steam (or coal) is easily doubled or trebled. Another point in connection with auxiliaries may be illustrated by an incident on board the *Charleston*. It had been found that considerable oil came from the dynamo engines to the feed tanks, but the cause was difficult to discover. Finally, one day an electrician was caught emptying a bucket of oil and water that he had taken from inside the engine casing by simply opening the drains to the L.P. cylinder of the engine and letting the vaccum draw it into the condenser. It is hardly necessary to add that this act of supreme laziness and gross ignorance was discontinued.

APPENDIX A.

PORT ROUTINE.

5·30 to 7·30 A. M.	8·15 to 11·30 A. M.	1·15 to 4·30 P. M.
Monday. Men who wish to, may scrub clothes until 6·00 A. M. Hoist ashes. Pump all bilges dry, clean stations, and work of day as ordered.	Move main engines and auxiliaries not in use. Work of the day. Test water in boilers in use and in feed tanks. Give boilers in use surface and bottom blow and test try cocks. Clean grease extractors. Blacksmith overhaul firing tools.	Finish uncompleted routine work and work of the day.
Tuesday. Same..................	Move main engines and auxiliaries not in use. Work of day. Move all cocks and valves not ordinarily moved. Overhaul steam traps.	Same.
Wednesday. Same..................	Move main engines and auxiliaries not in use. Overhaul W. T. doors and mechanical devices. Test deck operating gear of W. T. doors and of boiler stop and safety valves. Give boilers in use surface and bottom blow and test try cocks.	Half holiday if there is no urgent work.
Thursday. Same..................	Move main engines and auxiliaries not in use. Estimate coal on hand and inspect bunkers, scaling and painting when necessary. Work of day. Clean grease extractors.	Uncompleted routine work and work of the day.
Friday. Hoist ashes. Clean bilges and bilge strainers. Blacksmith overhaul machine shop tools and wrenches of Dept. Coppersmith inspect salt-water piping and repair oil cans, oil lights, etc.	Clean bilges, inspect D. B. compartments and renew gaskets of manholes when necessary. Test main and auxiliary drains by pumping out bilges, and test gear for operating main drain valves from deck above.	Same.
Saturday. Hoist ashes. Field day. Scrub paint work.	Move main engines and auxiliaries not in use. Scrub paint work and clean stations.	Half holiday.
Sunday. Hoist ashes, pump out bilges and clean stations.	Before quarters clean stations and prepare for Sunday inspection.	

NOTE.—Whenever a job of work is completed or work is discontinued for the day, all tools and stores must be returned to the storerooms and wrenches to wrench boards.

Every Day.—Before 7.50 P. M.—Storeroom keepers report unuseds tores, and tools returned to storerooms. Close all W. T. doors and manholes not to be used during the night. Take down all clothes hanging in unauthorized places. Sound D. bottoms and log fresh water on hand. *At 7.50 P. M.* report department secure.

Night Watches.—8.00 to 12.00 P. M.—Watch cleans all number 1 and 3 furnaces. *Mid. watch* sweeps tubes of boilers in use, except Saturday night. Morning watch cleans all number 2 and 4 fires.

SEA ROUTINE.

5.30 A. M.—Call working party, detail for trimming down coal, machine-shop men, cooks, washroom keepers and other men who do not stand watch.

6.00 A. M.—Special details and idlers go ahead with work of the day as per port routine and orders.

6.00 to 8.00 A. M.—Pump out engine room, fireroom and shaft-alley bilges and clean bilge strainers. Boilermakers inspect boilers, and coppersmith water piping for leaks. Send up to blacksmith firing tools that need overhauling.

8.00 to 9.00 A. M.—Warrant machinist on watch inspect engine and firerooms, and relief warrant machinist remainder of engineer's department and see same ready for inspection. Have clothing and shoes in unauthorized places stowed away and take dry clothes off clothes lines in engineer's department.

6.00 to 8.00 P. M.—Sound all double bottoms and pump bilge double-bottom compartments dry. Close all watertight doors and manholes which are not to be used during the night. Chief water tender of watch give boilers in use light bottom blow, and light surface blow if water is greasy. Test water in boilers in use and in feed tanks with hydrometer and by tasting. Blacksmith see coal bunkers are not overheated and obtain temperatures if possible.

7.50 P. M.—Warrant machinist of watch report department secure in accordance with orders and routine.

Each watch will clean one fire in each boiler and blow tubes in one-fourth of all boilers in use in regular rotation. At five bells hoist ashes. (4.00 to 6.00 P. M. watch does not hoist ashes and 6.00 to 8.00 watch does not clean fires.) Use test cocks frequently and blow through all gauge glasses at least once each watch. Before being relieved pump out bilges, lay out coal for next watch and sweep up floor plates.

ROUTINE ORDERS.

(To be kept at log desk in starboard engine room.)

No. 1. July 30th, '06.

Orders in this book will be copied in a smooth order book to be kept in the engineer's office, and the yeoman will see that each order is initialed in the smooth order book by all engineer officers and such petty officers as the senior engineer may direct.

All officers and petty officers will be responsible for the proper execution of these orders.

Officers or petty officers in charge of a watch will see that this order book is not lost or defaced.

(All officers and petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 2. July 30th, '06.

Attention is called to the port and sea routines posted in the bulletin board, which will be followed except when modified by the senior engineer.

(All officers and petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 3. Aug. 1st, '06.

When in port auxiliary watches will carry boiler steam pressure between 150 pounds and 175 pounds.

(All water tenders and petty officers who have charge of auxiliary watches initial.)

——— ———, *Sen. Eng. Off.*

No. 4. Aug. 1st, '06.

Whenever the ship is listed the list will be corrected by using coal and make-up-feed entirely *from the side towards which the ship is listed.* Boilers are not to be pumped down or up in order to correct the list.

(All water tenders and machinist's mates initial.)

——— ———, *Sen. Eng. Off.*

No. 5. Aug. 2d, '06.

Drinking-water buckets will not be used for any other purpose than drinking water; when not in use they will be turned into the storeroom, and storeroom keepers will see that they are kept properly marked "Drinking Water," in large letters with white paint.

(All water tenders, machinist's mates and storeroom keepers initial.)

——— ———, *Sen. Eng. Off.*

No. 6. Aug. 6th, '06.

Whenever water is taken out of or put into a reserve-feed double bottom, the fact will be noted in the log, thus: "Make-up-feed, B–99, 35 inches to 30 inches." "Filling boilers, B–98, 42 inches to 0 inches." "Distilling to B–94, 25 inches to 42 inches." "Water from water barge to B–95, 0 to 50 inches." It will be seen that this gives the reason why water is pumped in or out, and the soundings in inches both before and after pumping.

(All water tenders and machinist's mates initial.)

——— ———, *Sen. Eng. Off.*

No. 7. Aug. 10th, '06.

Hereafter there will be noted on the rough log sheet the amount of coal used for each watch from each bunker. The buckets of coal will be noted in this manner: "B–5, 10 buckets;" "B–7, 14 buckets;" "B–8, 22 buckets."

(All machinist's mates and water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 8. Aug. 10th, '06.

Whenever a coal bunker, double-bottom compartment, boiler or other confined space is opened, no man will be allowed to enter it until the air has been tested with a candle or bunker light. If the light goes out or burns dimly no one will be allowed to enter until the space is properly ventilated. Men at work in a confined space such as the above must be in constant communication with some one outside.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 9. Aug. 10th, '06.

When fires die out in any boiler the following work will be done as soon as possible:

(*a*) Give boiler light surface and bottom blow while pressure is on. Sweep tubes with steam hose. Open air cock when pressure is gone. Haul furnaces and ash pits, and overhaul grate surfaces. Renew any broken fittings. Overhaul gauge glasses and try cocks if they are out of order. Inspect

all manholes and handholes and seams, and stop leaks. Grind in leaky valves and remake leaky joints. See that dampers are working properly.

(*b*) If steam space is opened up examine zincs, and renew if one-half gone. Examine all interior piping and fittings which are accessible. Clean grease and scale off all surfaces.

(*c*) When through overhauling boiler, pump entirely full of fresh water, and close furnace and ash-pit doors. Test boiler with hydraulic pressure, if so instructed.

(All chief water tenders and boilermakers initial.

——— ———, *Sen. Eng. Off.*

No. 10. Aug. 10th, '06.

Whenever boilers, cylinders, feed tanks, condensers, etc., are to be entered for inspection or overhauling, all valves which would allow steam or water to enter must be secured by locking, lashing, or otherwise. Likewise, whenever anyone is working in a main engine cylinder, the turning engines will be secured.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 11. Aug. 12th, '06.

When raising steam in any boiler the following routine will be strictly followed:

(*a*) Pump boiler to steaming level.

(*b*) See no clothes or inflammable material around boiler or uptakes, and smoke-pipe covers off.

(*c*) See air cocks, steam-gauge valves and water-gauge valves are open.

(*d*) See all boiler-stop and surface and bottom-blow valves are closed.

(*e*) Start fires and report to engine room and officer of the deck that fires are started.

(*f*) See feed pump ready for use, and call a boilermaker who will look out for leaks and set up on joints in boiler if necessary.

(*g*) Report to engine room when steam forms.

(*h*) When steam has blown through air escape several minutes, close air cocks, test try cocks and blow through gauge glasses.

(*i*) Let fires burn slowly and evenly and do not force boilers without specific orders to do so.

(*k*) When pressure is about 100 pounds, try safety valves by lifting with hand gear.

(*l*) When steam pressure is a few pounds above that in the steam line, connect up boiler by opening up stop valve slowly.

(*m*) Report to engine room and officer of the deck when boilers are connected.

(All chief petty officers, water tenders and boilermakers initial.)

——— ———, *Sen. Eng. Off.*

No. 12. Aug. 10th, 06.

When turning steam into any steam line great care must be taken to avoid "water hammer." Drain valves on the steam line will be opened before admitting any steam, and the line will be heated up slowly by gradually opening the steam valve.

(All petty officers initial.)

——— ———, *Sen. Eng. Off.*

No. 13. Aug. 25th, '06.

At sea the yeoman will detail two men from each section, and water tender or other petty officer in charge, to throw down coal from the upper to lower bunkers. This coal-bunker detail will work during the day as per routine, and begin throwing down coal from the upper bunkers as soon as coal stops running out on the fireroom floor plates through lower bunker doors.

Bunker plates must not be left open when washing the upper decks or at night. When under forced draft care must be exercised that air does not blow through bunkers from the firerooms thus spoiling the draft and dirtying the upper decks.

(All water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 14. Aug. 25, 1906.

Water tenders in charge of a watch will, as soon as they come on watch, inspect all the firerooms to see that the routine and routine orders are being carried out, that fires are in proper shape, water at proper height in gauge glasses, feed pumps working properly, and everything about the boilers in good condition.

They will at sea allow no fires to be cleaned or tubes to be blown until there is over 175 pounds steam pressure, and then they will not allow too many fires to be cleaned at one time.

If necessary to keep steam pressure above 175 pounds, only one fire will be cleaned at one time.

Tubes will be properly blown and lances will be entered their full length between the nests of tubes.

(All water tenders initial.)

——— ———, *Sen. Eng. Off.*

No. 15. Aug. 25, 1906.

Clothing may be hung in the engineer's department only on clothes lines put up in the engine and firerooms by chief petty officers in charge of those compartments. Clothing left adrift will be turned into the master-at-arms.

(All petty officers initial.)

——— ———, *Sen Eng. Off.*

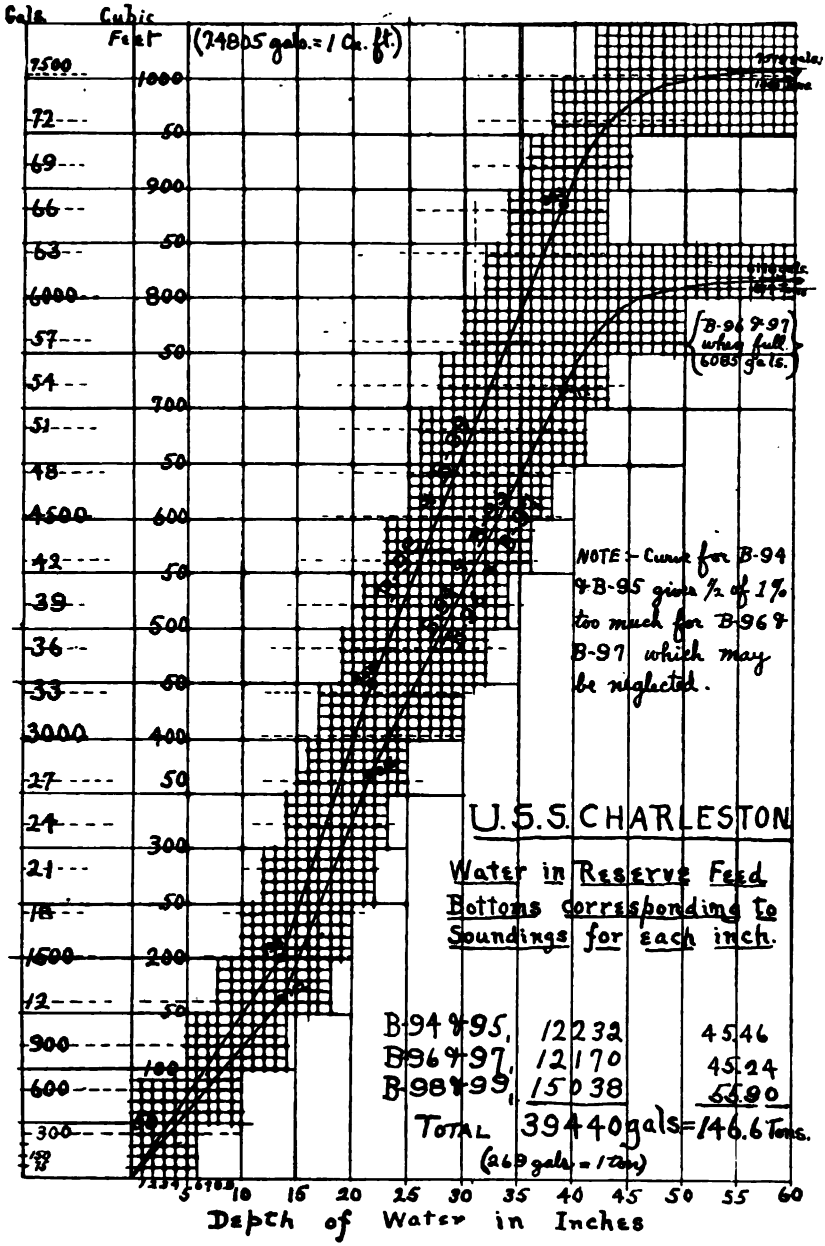

Gals.
Cubic Feet
(7.4805 gals. = 1 Cu. ft.)
{B-96 & 97 when full 6085 gals.}
NOTE — Curve for B-94 & B-95 gives ½ of 1% too much for B-96 & B-97 which may be neglected.
U.S.S. CHARLESTON
Water in Reserve Feed Bottoms corresponding to Soundings for each inch.
B-94 & 95, 12232 4546
B-96 & 97, 12170 4524
B-98 & 99, 15038 5590
TOTAL 39440 gals = 146.6 Tons.
(269 gals = 1 ton)
Depth of Water in Inches

Fig. 3.

U. S. S. *CHARLESTON.*

Capacity of Reserve-Feed Tanks, B-98 and B-99.

Soundings, in inches.	Cubic feet.	Gallons.	Soundings, in inches.	Cubic feet.	Gallons.
5	70	525	33	728	5,460
10	145	1,087	34	754	5,655
11	160	1,200	35	780	5,850
12	175	1,312	36	806	6,045
13	193	1,440	37	832	6,240
14	210	1,575	38	857	6,427
15	238	1,785	39	882	6,615
16	271	2,032	40	910	6,825
17	300	2,250	41	930	6,975
18	325	2,437	42	943	7,077
19	355	2,662	43	956	7,170
20	388	2,910	44	967	7,252
21	425	3,187	45	978	7,335
22	450	3,375	46	982	7,365
23	475	3,562	47	987	7,402
24	500	3,750	48	994	7,455
25	525	3,937	49	997	7,477
26	550	4,125	50	999	7,492
27	575	4,312	51	1,001	7,507
28	600	4,500	52	1,002	7,514
29	625	4,687	53	1,003	7,522
30	651	4,875	54	1,003½	7,526
31	677	5,077	55	1,004	7,530
32	703	5,272	56	1,005	7,537

269 gallons, equal to one ton (2,240 pounds).

Tons marked on D. B. manholes are not correct.

U S. S. *CHARLESTON.*

Methods of Pumping Out Compartments.

Coal bunkers, magazines and other compartments not shown in this table have no drains. They may be pumped out with a suction hose.

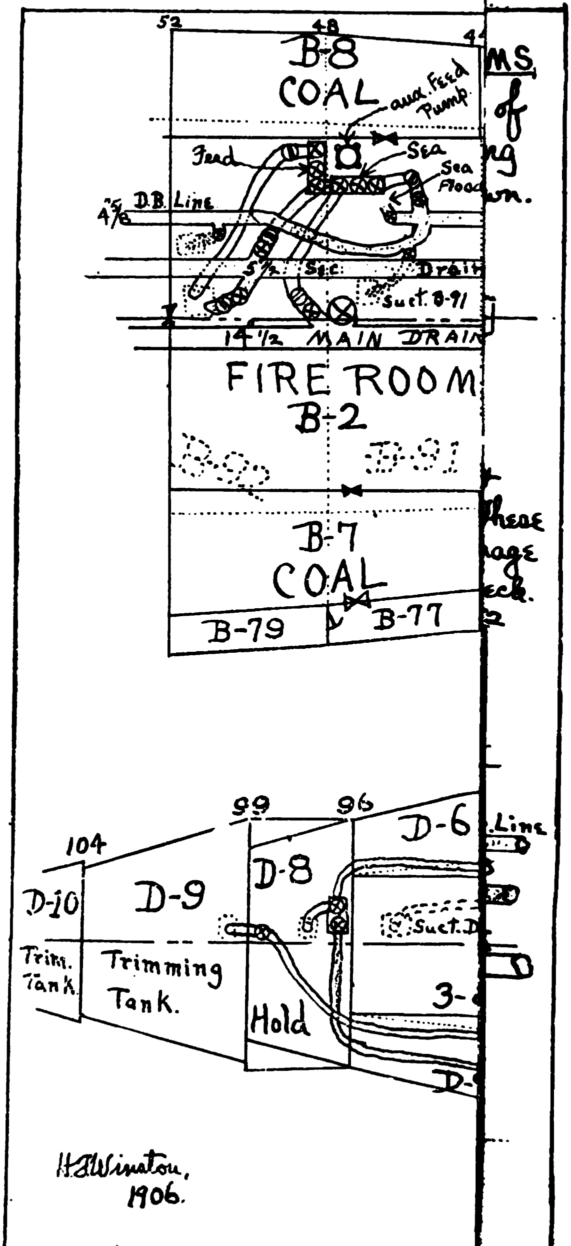
Compartments.	How drained.
A-1. Trimming tank......	Sluices to A-2.
A-2. Trimming tank...... A-3. (3 suctions.) Forward hold.	Through man. No. 4, and sec. drain cut-out valve (in No. 1 fireroom), to *secondary drain;* or through man. No. 4 and man. No. 10 to *hand pump.*
A-5. Lower dyn. room...	Same as A-2.
A-20. Upper dyn. room...	Sluices to A-5—starboard and port.
A-97. Ford. hold D. B....	Same as A-2.
A-98. D. B. of 3-in. mag..	Same as A-2.

Compartments.	How drained.
A-99. Dyn. room D. B....	Double-bottom suction line to *auxiliary feed pumps, Nos. 2 and 3.*
A-117 and 118 chain lockers. (2 suctions.)	Same as A-2.
B-1, 2, 3 and 4. Firerooms 1, 2, 3 and 4......	*Main or secondary drains* direct. Also direct bilge suction to *Auxiliary feed pump in each fireroom.* (Firerooms No. 1 and No. 2 sluice to D. B.'s under them.)
B-89, 90, 91, 92 and 93. Fireroom double bottoms.	*Auxiliary feed pumps No. 2 or No. 3* on double-bottom suction line. (There is a cut-out valve in suction line to auxiliary pump No. 2.)
B-94, 95, 96, 97, 98 and 99.	Reserve-feed tanks and connect to feed pumps.
B-109. Evaporator room..	Sluices to fireroom No. 4, starboard and port.
C-1. Starboard engine room.	*Stbd. M. circ. pump* direct bilge injection. *Main drain direct. Secondary drain direct* or on stbd. eng.-R. bilge line. *Stbd. F. and B. pump* or *shaft pump* on engine-room bilge line direct to bilge or crank pits. (There are sluices to D. B.'s under engine room.)
C-2. Port engine room...	Same as for starboard side.
C-97, 98 and 99. Engine-room double bottoms.	*Sec. drain* through man. No. 1 in port engine room and double-bottom line. *Port F. and B. pump* through man. No. 1 and double-bottom main. *Port shaft bilge pump* or *port F. and B. pump* through port engine-room bilge line, man. No. 1 and double-bottom suction line.
D-1. M. 3-in. magazine..	No drain.
D-2. Port shaft alley. (2 suctions.)	*Secondary drain* through man. No. 1 in port eng. room (cut-out valve in sec. drain near manifold) and port shaft-alley manifold in after part engine room. *Port shaft pump* or *port F. and B. pump* through port engine-room bilge line and shaft-alley man. (Cut-out valve on branch to engine-room bilge line.)
D-3. Stbd. shaft alley.....	*Sec. drain* through man. No. 2 in stbd. engine room and stbd. shaft-alley manifold aft in engine room. *Stbd. F. and B. pump* or *shaft pump* through shaft-alley manifold and stbd. engine-room bilge line. (Cut-out valve in branch to engine-room bilge line.)
D-4 and 5. Shaft alley wing compartments.....	No drains.
D-6. Wing D. Bottom...	*Sec. drain* through man. No. 3 (aft in port engine room), after bilge line and man. No. 5 in port shaft-alley. *After hand pump* through hand-pump man. (in P. eng. R.) man. No. 3 and man. No. 5, as above.

Compartments.	How drained.
D-7. Wing D. Bottom...	*Sec. drain* through man. No. 3 (aft in P. eng. R.), after bilge line and man. No. 6 in after hold. *After hand pump* through hand-pump manifold and manifold No. 3.
D-8. After hold..............	Same as D-7.
D-9. Trimming tank......	*Sec. drain* through man. No. 3 (P. eng. R.) or *after hand pump* through pump manifold and man. No. 3.
D-10. Trimming tanks...	Sluices to D-9.
D-98 and 99. 3-in. magazine double bottoms.	Same as D-6.
D-25 and 26. Steering engine flats.	Sluice to D-8. (After hold.)

Pump Connections, Main Drain.

Pumps.	How connected to main drain.
Starboard main circulating.	Open main-drain suction and overboard discharge at pump.
Port main circulating......	Same as starboard.
No. 1, auxiliary feed......	Suction valve at each pump and cut-out valve in branch to main drain in each fireroom.
No. 2, No. 3 and No. 4, auxiliary feed.	Same as No. 1 auxiliary feed.
Starboard engine-room fire and bilge.	Man. No. 2 abaft pump and cut-out valve in branch to main drain (in bilge ford. in starboard engine room).
Port eng.-room fire and bilge.	Man. No. 1 abaft pump and cut-out valve in starboard engine room as above.
Starboard shaft pump.....	Starboard engine-room bilge line, man. No. 2 and cut-out valve as above.
Port shaft pump............	Port engine-room bilge line, man. No. 1 and cut-out valve as above.
After hand pump............	Bilge suction of after hand pump man. (in P. engine room), man. No. 3 (aft in port engine room), man. No. 1 (in port engine room) and cut-out valve as above.



52
48
B-8
COAL
aux. Feed Pump
Feed
Sea
Sea Flood
D.B. Line
4 7/8
Suc.
Drain
Suct. B-91
14 1/2 MAIN DRAIN
FIRE ROOM
B-2
B-92
B-91
B-7
COAL
B-79
B-77
104
99
96
D-6
D-10
Trim. Tank
D-9
Trimming Tank
D-8
Hold
Suct. D
3-
H.G. Winston,
1906.

Pumps.	How connected to main drain.
Dist. circ. pump............	Close sea valve at ship's side (stbd. eng. room), open "sea" valve in man. No. 2 (stbd. engine room), main drain valve in man. No. 2 and cut-out valve as above.

Pump Connections, Secondary Drain.

Secondary drain connects to main drain through a branch in starboard engine room (branch has a cut-out valve) and manifolds No. 1 or No. 2. Consequently main drain pumps may be used in this way on the secondary drain.

Pumps.	How connected to secondary drain.
No. 1, No. 2, No. 3 and No. 4 aux. feed pumps..	Suction valve at each pump and cut-out valve on branch to secondary drain in each fireroom.
Forward hand pump.......	Hand pump man. in No. 1 fireroom, man. No. 10 in same fireroom and cut-out valve to secondary drain (just abaft man. No. 10).
After hand pump...........	Hand pump man. (in port engine room), hand pump suction in man. No. 3 (port engine room) and secondary drain valve in man. No. 3.
F. and B. pump, starb'd engine room.	Direct suction through man. No. 2 in starboard engine room, or through engine-room bilge line and man. No. 2.
F. and B. pump, port engine room.	Same as above, except man. No. 1 in port engine room.
Starboard shaft pump.....	Starboard engine-room bilge line and man. No. 2 in starboard engine room.
Port shaft pump............	Port engine-room bilge line and man. No. 2 in port engine room.
Distiller circ. pump........	Close sea valve at ship's side in starboard engine room. Open "sea" valve in man. No. 2 and secondary drain valve in same manifold.

APPENDIX B.

DUTIES OF YEOMAN AND ASSISTANTS.

Daily, before quarters, correct lists of absentees, sick, working parties, etc., in engine room, and at same time take down obsolete lists and notices.

Bring to quarters lists of excused absentees.

Before noon make up coal report and fill in "bunker expenditure book;" write up and have signed log book

(reading water meters to see that gallons of water distilled "for feed" and "for ship" are correct, and read scale on gauge glasses of oil tanks for correct amount of oil "expended" and "on hand").

About 11·30 A. M. set all clocks of engineer's department, obtaining correct time from chief quartermaster.

At noon take log and coal account to senior engineer's room for his signature.

At 7·30 P. M. take "daily order book" to chief for orders of the next day.

At end of week enter up expenditures of stores as per rough books of storeroom keepers. Bring up to date smooth copy of log to be sent to Washington.

At end of month correct and bring up to date watch, quarter and station bills and all bulletin-board notices. General straightening up of everything in engineer's office. Submit to captain monthly report of expenditures and receipts of stores.

Quarterly send in smooth copy of steam log and quarterly reports (page 381, Navy Reg.) to Navy Department. Make out surveys for stores, Title "B," which are lost or damaged.

Semi-annually (June and December), make out requisitions for six months' supplies, Title "Y," and such articles as have been condemned by survey under Title "B." Correct "spare-part book" and requisition for spare parts used up. (This book contains a list of the spare parts of machinery on board and where each is stowed.)

Annually (December,) take annual inventory of stores.

Requisitions for stores on allowance book, of which the amounts are "as required," such as oil, waste, grate bars, etc., must be made out when they run low.

Books of supplies and accounts will be kept as per Art. 1546, Navy Regulations.

Copies of all official letters written will be kept in a book marked "Letters Sent," and originals or copies of all letters received will be kept in a book marked "Letters Received."

The "Coal-bunker expenditure book" will be made out in accordance with the following form:

Lower bunkers...... / Upper bunkers......	B–5 / B–101	B–6 / B–102	B–7 / B–105	B–8 / B–106	B–9 / B–111	B–10 / B–110	B–11 / B–115	B–12 / B–114	Totals.
	79	79	112	112	123	123	126	126	
	103	103	100	100	102	102	100	100	
Capacities	182	182	212	212	225	225	226	226	1690
Sept. 30, on hand..	182	182	212	212	225	225	226	226	1690
Oct. 1. Expended..	3^{220}	...	...	...	19^{80}	24^{320}	30	...	76^{870}
Remaining	178^{2020}	182	212	212	205^{2210}	200^{1920}	196	226	1614^{1670}

A rough note book marked "Repairs Needed," will be kept hanging in a conspicuous place, in which officers of the department may note desirable repair work, and from which "necessary repairs" may be submitted to the captain on arrival in port.

It will be noted that at the top are the lower bunkers with the bunkers which trim into them just below. Then come the capacities of the bunkers and of each pair of bunkers. In making up the noon coal report it is usual to add together the expenditures for each watch during the preceding 24 hours. The coal-bunker book is filled in by entering the expenditures from each bunker (or pair, upper and lower) for the preceding 24 hours. Then the "total remaining," in last column, should agree with the noon coal report and is a check on it.

By keeping this separate coal account for the various bunkers there is always on hand a check for the amounts estimated in each bunker by inspection. In addition, whenever a bunker is emptied the bunker book shows at once if the proper amount of coal is being expended each day. If not, either the number of buckets turned in is incorrect or the number of pounds allowed per bucket is wrong, and the error may be corrected before there is a large discrepancy.

Delicate instruments, such as drawing instruments, micrometer gauges, indicators, gauge-test outfits, etc., will be kept in the log room, and the yeoman will be responsible that they are so stowed that they will not be broken.

Drawings will be taken from the log room only by an officer, and he will place his initials opposite the number of the drawing in the "Index Book" of drawings when the drawing is taken out. Engineer's pocket books and other books designated will not be taken from the office.

Text books may be taken by any member of the engineer's force for one week, and will be charged by the yeoman when taken out and the date noted.

Keys from the key board will be delivered only to an officer or person who is responsible for the compartment or mechanical device to which the key belongs.

The log room will be locked and the key kept by yeoman, or turned in to the senior engineer officer when none of the yeoman force are there.

When getting under way or coming to anchor there will be a yeoman or one of his assistants at each engine-room desk to make notes for the log.

DUTIES OF STOREROOM KEEPERS.

The senior or head storeroom keeper has general charge of all the storerooms and superintends the stowage of stores and their issue, and is responsible for the care, preservation and proper expenditure of stores.

All storeroom keepers will stand watch with their sections, or as the senior engineer may direct, and will observe the rules prescribed for storeroom keepers.

The senior will keep all storeroom keys and see all storerooms locked whenever no storeroom keeper is on watch. In case of his absence from the ship these keys will be turned over to the next storeroom keeper in seniority or to the engineer's yeoman.

Stowage.—As far as possible all stores of the same kind will be stowed together and in such a manner that a portion of each kind of stores are easily seen. Stores of one kind must never be entirely hidden by stowing them behind other stores. Articles wrapped up in packages, or in boxes, must be properly marked or tagged, with the marking in plain view.

Articles of steel or iron must not be stowed in damp places where they will rust.

Rubber goods and composition packings must not be stowed in a hot, dry place, where the rubber soon drieds up and becomes brittle.

All instruments and other articles which may be easily broken (steam gauges, thermometers, etc.) must be stowed carefully where they will not be injured when the ship is at sea.

Spare parts of machinery, such as heavy bolts and nuts of steel, will be painted with white lead to prevent rusting. Other spare parts of steel or iron will be covered with white lead and tallow or vaseline (when bearing surfaces). All spare parts likely to be injured will be wrapped with canvas in order to protect them.

Once a quarter (in March, June, September and December) all spare parts will be examined and overhauled if necessary. Whenever spare parts are used or their stowage changed the yeoman is to be notified so he may correct his "spare-part book."

In December the annual inventory of stores will be taken.

Books.—A charge book for tools, instruments, etc., will be kept in each storeroom, and such articles will be charged against the person taking them out. When the day's work is completed these tools, etc., must be returned to the storerooms, and before 8·00 P. M. each day the storeroom keeper on watch will report to the officer on duty whether all are returned or not.

No tools or stores will be issued to any one outside of the engineer's department, except by order of the senior engineer.

In a rough expenditure book will be entered the stores expended each day, and at the end of the week this book will be turned in to the yeoman, who will copy the expenditures into his book.

There will also be kept a list of stores, Title "B," which, having been used, damaged or lost, must be surveyed.

Stores such as oil, waste, paint, packing, canvas, etc. (which are in Title "Y"), will be issued only upon the order of an officer of the department.

Drinking-water buckets will be served out when making preparations for going to sea, and returned to storerooms upon arrival in port.

The storeroom keeper will see that they are properly painted a light blue and lettered in large white letters "Drinking Water," before serving them out. All wash buckets will be painted with a broad green stripe before being served out.

Men who come to storerooms for stores must wait outside, and have same served out to them by the storeroom keeper.

Wash-Room Rules.—Each wash-room keeper will give the paint work, decks, bowls, etc., in his wash room a thorough scrubbing each day before morning quarters. All men must be clear of the wash room twenty minutes before quarters so it may be properly cleaned for inspection.

Wash rooms will also cleaned up after dinner and after 8·00 P. M., as soon as the watch finishes washing up.

Wet clothing will not be allowed to be hung in wash rooms.

When leaks are discovered in the water piping, or drain pipes are plugged, the plumber will be notified, and if he does not repair the defects, the fact will be reported to the officer on duty. Anyone who fails to obey the above rules will be reported to the officer on duty in the engine room.

Rules for Steam Launches.—Whenever the order is received from the officer of the deck to secure the launch, the boiler will be given a light blow by opening the blow-down valve wide and then closing it again immediately. At the same time the boiler tubes will be swept with a steam lance, if there is no fresh paint work near the launch, which will be soiled.

When raising steam the safety valve will be lifted by hand to see that it does not stick, and water gauges blown through and try cocks tested.

If it is necessary to feed salt water into the boiler at any time the fact will be reported to the officer in charge of auxiliaries as soon as possible, and the boiler will be opened up and cleaned shortly thereafter.

Once a quarter (in January, April, July and October) all

the launch machinery will be opened up, examined and overhauled. This includes the interior of boiler, cylinders and valve chests of engine, air pump, feed pump, water tanks, coal bunkers, bilges, and supports for engines and boilers, and the keel condenser. The settings of valves and clearances of pistons will be taken, but will not be altered except under the direction of officer in charge of auxiliaries.

Whenever the launch is to remain in its cradle for five days or more the cylinders will be opened up, wiped out and lightly coated with vaseline.

Spare parts of machinery in the launch will be kept covered with white lead and tallow or vaseline, and examined frequently to see that they are not rusting. When spare parts are used the fact will be reported to the yoeman so his books may be kept straight.

Machine-shop Rules.—The machinist in charge will be responsible for the good condition of the machine shop and the care and preservation of all machines and tools there.

When he and his helper (fireman or coal passer) are absent from the machine shop he will lock the door and keep the key, or, if he is to leave the ship, turn it in to the engineer's office. He will keep a charge book or slate and charge up tools to the person taking them out. If these tools are not returned by 8·00 P. M. he will report the fact to the officer on duty.

No one will be allowed to stow clothes or other articles in the machine shop at any time except the machinist in charge and his helper, who will be allowed to stow personal belongings only in such space as may be assigned them.

A list will be taken of all stores or tools which have been expended, lost, damaged or worn out, and handed into the yeoman each Saturday morning to be entered on his books.

Rules for Evaporator Room.—Whenever any evaporator is started or shut down the fact will be reported to the engine room.

In starting up an evaporator, water will be discharged to the

reserve-feed tanks until it is known by testing and tasting the water that it is perfectly fresh.

When running an evaporator, the water level will always be low enough to leave the upper row of the nest of steam tubes bare. Shell pressure will be kept at from 6 to 8 pounds, and should be even lower when starting up an evaporator that has just been scaled.

Every four hours, at seven bells, the evaporator in use will be blown down, and when empty will be quickly filled to steaming level with cold water before admitting any steam. Then the steam valve will be opened and steam rapidly admitted to the nest of tubes. This quick cooling and then heating of the tubes causes them to first contract and then expand, and cracks off the scale.

Great care must be taken that idle evaporators are kept dry to prevent rapid corrosion and rusting. For this reason particularly, all valves which would allow water or vapor to back up into the shell must be kept closed tightly, and these valves renewed or ground in as soon as they begin to leak.

After six weeks' continuous use evaporators will be scaled and interior of shell thoroughly examined. Scale need not be removed from the shell, as it protects the surface of the metal. Interior piping must be examined to see that it is not plugged up or corroded, and the slits in dry pipe freed of any deposit.

APPENDIX C.

MONTHLY ROUTINE.

Obtain some indicator cards of main engines.

Senior engineer inspect all compartments, watertight doors and mechanical devices and submit written report to captain.

Inspect auxiliary steam machinery under other bureaus.

Clean out feed and filter tanks and wash or renew filtering material.

Test fireroom blowers.

Scale evaporators that have been in use and renew zincs, if one-half gone.

Inspect steam-launch machinery. (Interior of boiler, safety valves, check valves, pumps, engine, shafting and keel condenser.)

Overhaul oil service of main engines.

QUARTERLY ROUTINE.

Examine holding-down bolts and all nuts about the framing of main engines.

Examine valves of main air pumps, of auxiliary air and circulating pumps, and the steam cylinders and valves of these pumps.

Boil out main and auxiliary condensers with soda.

Open up main-engine cylinders, examine piston rings and springs, and measure clearances.

Open up main-engine valve chests, examine piston rings and clean balance pistons.

Try relief valves on main cylinders and jackets and overhaul jacket reducing valves.

Paint engine and firerooms.

Inspect and overhaul all spare parts of machinery, tools and wrenches.

Test steam gauges on engine-room gauge boards, of boilers and important gauges of auxiliary machinery and steam lines.

Examine main shafting, bearings, stuffing boxes and propellers as far as possible.

Examine zincs in zinc boxes of salt-water piping, and renew if one-half eaten up.

Overhaul boiler-feed and hotwell pumps.

Test safety, relief and reducing valves of auxiliary steam piping and auxiliaries of department.

Open up boilers, clean interior parts, examine interior fittings, and renew zincs if one-half gone.

Set safety and sentinel valves of boilers if not correctly set.

Clean air space between uptake and boiler casing.

Overhaul uptake dampers.

SEMI-ANNUAL ROUTINE.

Open up all condensers and examine tubes.

Examine shafting and interior of water chamber of main circulating pumps.

Examine valves in all manifolds.

Go over main and auxiliary exhaust piping and remake joints where leaking.

Grind in or renew valves in steam lines which ordinarily cannot be taken out on account of constant use.

ANNUAL ROUTINE.

Test compressed-air system to full pressure.

Take yearly inventory of stores.

SPECIAL OCCASIONS.

Upon Arrival in Port.—Report of repairs necessary (Art. 669, *Navy Reg.*).

Obtain coal, oil, water and waste, if needed.

Very Cold Weather.—Keep piping in steam launches from freezing. Either keep steam up and turn over engines frequently or drain the machinery entirely, including the keel condenser.

In Dry Dock.—Arts. 1638–9, Navy Reg.

Dock Trials.—Art. 1601, Navy Reg.

Speed Trials.—Arts. 1602 to 1605, Navy Reg.

BOOKS AND NOTES FOR REFERENCE.

Barton's "Naval Engines and Machinery."

Bieg's "Naval Boilers."

Liversidge's "Engine Room Practice."

"Engineer's Pocket Book." (Kent's is a good one.)

"Carpenter's "Experimental Engineering."

Durand's "Practical Marine Engineering."

Neilson's "Steam Turbine."

Welsh's "Ship and General Painter's Hand-book."

(The above usually found in library on board ship.)

Bureau of C. and R. "Specifications" for the ship.

Bureau of C. and R. "Booklets of Plans and General Information."

Typewritten description of drainage and fire systems.

(Usually found in executive officer's office.)

Bureau of S. E. "Machinery Specifications."

Bureau of S. E. "Booklets of Pipe Plans."

"Synopsis of Official Four-hours' Trial" of ship.

"Speed, Power and Revolution Curves." (Standardization trial.)

(Usually found in engineer's office.)

The following are not found on board ship, but may be obtained from files of the "JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS," or, in the case of patented devices, generally by simply writing the manufacturers:

Babcock & Wilcox Company's "Marine Steam."

W. W. White's "Coal Consumption of Main and Auxiliary Machinery of the U. S. S. *Minneapolis*." (JOURNAL of A. S. N. E., Vol. X, No. 1.)

B. C. Bryan's "Method of Calculating the Coal Endurance and Steaming Radius of War Vessels Under Various Conditions." (JOURNAL A. S. N. E., Vol. XVII, No. 3.)

H. T. Winston's "Selection and Use of Certain Apparatus and Material." (JOURNAL A. S. N. E., Vol. XVII, No. 2.)

Description of ship and her trials as written up in the JOURNAL of A. S. N. E.

Description and notes on the care and management of special, patented machinery on board; such as air compressors, ice machines, reducing valves, governor valves, steam traps, anchor engines, steering engines, speed indicators, boiler time-firing devices, engine indicators, machine-shop tools and machines, etc. For example, the Leslie Company have issued "Instruction Book, No. 2" (for their reducing valves), and the Crosby Steam Gauge and Valve Company, "Practical Instructions for using the Steam Engine

Indicator." A catalogue of a dealer in engineering sorets will also be found useful, such as Montgomery Ward & Company's (New York) "Tool Catalogue."

U. S. S. *CHARLESTON.*

Theoretical Coal and Water Consumption with Natural Draft.

Sept. 19th, 1906. H. T. WINSTON.

Revs. of engines.....	70	80	90	100	110	120	130	From trial-trip curves.
I.H.P. of engines....	3,000	4,300	6,100	8,400	11,200	14,500	18,500	
Speed in knots.......	12	13.6	15.1	16.6	18.1	19.45	20.6	
Tons coal per day...	94	140	193	270	365	470	...	3 lbs. per I.H.P.
Tons coal per day...	73	103	140	188	248	321	410	2 lbs. per I.H.P.
Tons coal per day... (See note.)	68	96	133	183	234	303	388	1.8 lbs. per I.H.P.
Tons water per day. ¼ of coal (2 lbs. per I.H.P.)	18	26	35	47	62	80	102¼	

NOTE.—First entry shows coal per day at rate of 3 pounds per I.H.P. of main engines per hour plus 10 tons per day for auxiliaries. This is approximately the amount most vessels of the U. S. Navy burn.

The second coal entry shows the coal per day, assuming 2 pounds per I.H.P. per hour plus 10 tons for auxiliaries per day. This is a good showing, but should be expected with good coal and normal conditions.

The third coal entry shows coal per day at the rate of 1.8 pounds per I.H.P. plus 10 tons per day. It is probable that the coal consumption will always be greater than this, and a smaller coal consumption will indicate that the coal account is incorrect.

Water expenditure in tons per day is the extra water required by the boilers when using 2 pounds of coal per I.H.P. per hour (normal consumption).

CORROSION OF PROPELLER SHAFT, U. S. S. *RHODE ISLAND.*

By HENRY E. RHOADES, PASSED ASSISTANT ENGINEER, U. S. N., MEMBER.

These notes and the accompanying drawing, Plate I, are submitted, not because the writer assumes that he has made a great discovery, nor to exploit what he does (not) know about electrical science, but that (possibly) what may be presented here will give reflection and study upon a condition disclosed on one of the outboard parts of the steam machinery of one of the new warships, and the further investigation of which may prove of value and benefit to those who take interest in it.

The incident referred to is that of the starboard main propeller shaft of the battleship *Rhode Island*, recently built by the Fore River Shipbuilding Company, and its condition was discovered and reported upon by a special Board of officers appointed to inspect the ship while in dry dock at the Navy Yard, Boston, Mass., just after arrival from the works of the builders to be transferred to the Government, a little more than a year ago, in October, 1905. The Board was not able at that time to make the most thorough examination of the shaft, such as was recently made, hence the first impression was that the groove, shown by the spiral line, might have been caused by ordinary corrosion, or by a scratch by some obstruction during the dock trial. The more minute examination subsequently made showed that the groove was distinctly cut for a length of about 56 inches, extending in the form of a right-hand spiral, from a point a foot from the fairwater on the propeller strut forward to about 4 feet 4 inches from the fairwater, and running about two-thirds the circumference

around the shaft. This groove was about 1½ inches in width at the widest point, and its depth ranged from ⅛ to $\frac{15}{32}$ inch throughout its length, the deepest point being where the groove was the widest.

In addition to the groove, and indicated by the smaller spots on the print both forward and abaft of the spiral groove, there were counted 292 distinct pittings extending in a direct line with the axis of the shaft and within a surface about two-thirds the circumference of the shaft, so that fully one-fourth, or more, of the circumference of the shaft remained scatheless. Fifty-one of these pits were ⅛ to ¼ inch deep, and thirteen from ¼ to ⅜ inch deep, while the others were less than ⅛ inch deep. The pits were generally irregular in form, although many of them were cup-shaped or dished. As the shaft was 17 inches in diameter it will be observed that these pittings were confined to about two-thirds the circumference, none being outside of the fore-and-aft boundaries and all within the limit of the two ends of the spiral groove. The estimate made by the Board as to the loss of strength from the original shaft through the decomposition that had taken place prior to the ship being turned over to the Government, resulted in a recommendation that the contractors substitute a new shaft, to which proposition they demurred, upon the claim that the designed strength of the shaft as originally called for had not been diminished. The Bureau of Steam Engineering sustained the recommendation of the Board, however, and a new shaft was supplied and installed in the ship in December, 1906. It must be admitted that this action of the officers of the Navy was justified by all the conditions met, and that the Government has obtained no more than what was its just due.

A casual glance of the groove after the shaft had been removed suggested that it might have originated from carelessness while being forged; that it might have been due to the one in charge of the drop-hammer allowing his scaler, an iron rod, to which is attached the spatula for keeping the scale removed from the surface of the metal, to be caught

under the hammer and welded with the steel. The rod being of iron and the shaft of steel, the action of salt water against the different densities of metal would excite corrosion and in the due course of time the rod would become dislodged. But a more careful inspection of the groove, with the aid of a microscope, disclosed distinctly lines such as would be made by a line of twisted wires of, perhaps, three-quarters of an inch in diameter.

A further investigation of the groove and the pittings of the shaft led me to the conclusion that they were produced by electrolysis, that process of galvanic action which is one of the conditions of electro-chemical decomposition. This galvanic action was caused, no doubt, by bringing into contact two dissimilar metals—the steel shaft and the strands of copper wire—the action being further excited by the action of the salt water under the wire around the shaft. When the vessel was put into dry dock, the hypothetical wire rope had disappeared, it probably having become dislodged from its long resting place by the revolution of the shaft when steaming from the Fore River Iron Works, at Quincy, to the Navy Yard, Boston; but the distinctively marked outlines of the strands of the wire rope in the corroded groove were sufficient evidence that a length of copper-wire rope, presumably electric wire, had accidently, or carelessly, been dropped overboard and lodged itself over the shaft where it had remained for months. The length of the spiral gutter or groove in the shaft indicated that both ends of the wire rope had hung downward from the shaft and, possibly, grounded on the bottom.

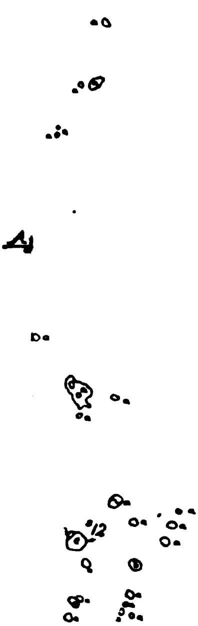
It was apparent that that part of the shaft most seriously pitted was that uppermost, or nearest the surface of the water, while the decomposition was in progress. By the contact of the two dissimilar metals—the steel shaft and the copper wire—they were charged with different electricities. The series of discharges that may be transmitted in a wire connecting a prime conductor (in this instance the steel shaft) of a machine in action with the ground possess the characteristics of a galvanic current, and these discharges were likely intensified

by the large mass of steel together with the copper-wire rope, and, probably, further excited by the composition sleeve on a part of the shaft.

It is a well-known fact that chlorine, which is the negative element of sea water, in itself decomposes metals with which it may be brought into contact; and the chemical energy of that liquid in contact with the dissimilar metals referred to, facilitated the discharge, and the current being thus maintained the electro-chemical decomposition was promoted. It is possible, also, that this decomposition, or electrolysis, may have been helped by the large percentage of manganese in the steel shaft; but no borings were taken, hence no determination was reached upon this proposition.

Whether or not these assumptions are correct, I concluded that the incident was so unusual and remarkable as to be well worth exploiting, because it may prove a text for further investigation of similar possibilities in the handling of warships.

inch to ½ inch.



HODE ISLAND."

STANDARD LIGHTNING PROTECTION FOR THE CONSOLIDATED POWER-PLANT CHIMNEYS AT UNITED STATES NAVY YARDS.

By N. Monroe Hopkins, Ph. D.,

Electrical Engineer for Consolidated Power Plants, Department of the Navy.

In the design of a modern central light and power plant for military purposes reliability of operation is the first essential.

There are many factors and influences, great and small, which tend to interrupt service, and they should all be carefully taken into account by the designing engineer. A breakdown of one of the new central power stations, or even a temporary failure to supply the important shops of construction and repair, equipment, ordnance and engineering, or the great electric pumps for emptying the dry docks, might, at a critical time, mean the defeat of a fleet. Insomuch as minute attention is given to the design of the machinery in duplicate and in triplicate, according to the best and most approved engineering practice, the work would be incomplete if the buildings and chimneys were not given the best protection from violent lightning strokes that modern practice can afford.

It has been with the view of providing chimney protection commensurate with the importance of the power plant to the naval service, that the writer took up the design of the lightning protection herein described. There are few devices in the field of applied electricity or electrical engineering regarded with greater skepticism and distrust than the lightning rod, but unfortunately so, for its reputation has been based in the majority of cases upon the old incorrectly applied con-

ductors of Benjamin Franklin's day. The early rods were put up with the idea that a lightning stroke took the form of a direct-current flash to earth, that it was attracted and influenced by the presence of iron or steel, and incapable of turning sharp corners or passing through glass insulators. Such were the old-fashioned views upon the character and behavior of electrical discharges. As the result of carefully conducted researches upon lightning strokes and phenomena, both in this country and abroad, the old Franklin rod has undergone a radical evolution and the value of correctly designed and erected lightning conductors has been completely demonstrated. According to the latest views, lightning does not strike a direct blow to earth, possessing inertia-like qualities which prevent it from turning sharp corners. The belief in its inability to pass through glass insulators has been abandoned also, and the danger of using insulators, for other reasons, has been fully demonstrated. The design illustrated and described in the present paper was completed after a careful study of recent English and Continental systems, as well as American practice. The writer has, in addition, tested several of the latest theories by practice and experiment, and reproduces herein a few of the typical experiments with apparatus capable of giving high electrical pressures at very high frequencies. In dealing with powerful oscillatory electrical discharges many complex factors must be considered in what is generally termed the circuit. The mathematical laws governing the deportment of direct currents are wholly inadequate where lightning discharges of oscillatory character are concerned, and it has been in connection with the study of proper protection for tall power-plant chimneys that some remarkable phenomena have been noted. It is not generally known or appreciated that the column of hot smoke and gases within a tall chimney conducts as well and at times better, than the metal rods outside. This point has been taken into account in the design of the lightning equipment for the power plants of the Navy. The modern theory of lightning protection is one of quiet equalization or neutrali-

zation of unlike electrical charges, as outlined in the following extract from an earlier treatise by the present writer:

*"Conductors are looked upon by many as a source of danger, they holding that the presence of a metallic rod so prominently located enhances electrical attraction, drawing bolts from the clouds which it is seldom capable of parrying or safely conducting to earth when struck. Although the old conductor of Franklin, but partially correct in application, carefully insulated from the building to which it was attached, occasionally, or we may, perhaps, say frequently, failed to afford protection when actually struck by lightning, it may be safely asserted that thousands of buildings have been silently shielded from demolition and the dwellers unconsciously protected, through the silent equalizing action of the old rod depending upon the effect of static electrical induction. This electrical phenomenon, but imperfectly anticipated by the old inventor philosopher, yearly caused his rod to shield these buildings without receiving credit, because of the silent and, apart from the glow of blue light at the rod tips, undemonstrative action. Franklin's rod was, and is yet, carefully kept from contact with the building by mounting it upon glass insulators, overtopping gables and chimneys for the purpose of receiving a lightning stroke should an electrical cloud come dangerously near, and of offering it a metallic path to earth, which it would follow in preference to that offered by the material of the building.

"The correct and true rod should not be erected to actually receive a lightning stroke, but to prevent any flash from taking place by a quiet, equalizing action."

It appears from a careful study of lightning strokes that there are two classes of discharge, either of them being capable of demolishing a chimney if not properly provided for. The following is based upon Sir Oliver Lodge's views on electrical storms:

"It has been pointed out by Sir Oliver Lodge† that light-

*"Approved Lightning Protection." Hopkins, Nevil Monroe. Ill. "Sci. Am. Supp." No. 1212, p. 19,434, March 25, 1899.
† Lightning Research Committee report of April 10th, 1905. Great Britain.

ning discharges are of two distinct characters, which he has named the A and B flash, respectively. The A flash is of the simple type which arises when an electrically-charged cloud approaches the surface of the earth without an intermediate cloud intervening, and under these conditions the ordinary type of lightning conductor acts in two ways: first, by silent discharge; and secondly, by absorbing the energy of a disruptive discharge. In the second type, B, where another cloud intervenes between the cloud carrying the primary charge and the earth, the two clouds practically form a condenser; and when a discharge from the first takes place into the second the free charge on the earth side of the lower cloud is suddenly relieved, and the disruptive discharge from the latter to the earth takes such an erratic course *that no series of lightning conductors of the hitherto recognized type suffice to protect the building*." The difference in character between the A flash and B flash, as named and described by Sir Oliver Lodge, may be more fully appreciated by referring to the accompanying illustrations. Here we have depicted a power plant, with its tall chimney subjected to the two kinds of lightning strain and stroke. In figure 1 at A we have before the actual stroke takes place, if it takes place at all, an inductive effect, and quiet, equalizing brush discharge, which may, if a sufficient number of sharp points are provided, actually prevent the disruptive discharge or stroke taking place. The A class may be looked upon as producing a steady strain, which may be quietly taken care of without damage to the chimney, if provided with a correctly designed and applied system of conductors. In the three following illustrations conditions are represented which give rise to the B class of strain and stroke, that which is far more difficult to safely handle by any system of conductors. In figure 2, an A type of flash between the two clouds precipitates a B type of flash between the lower cloud and the chimney top. In this case, the lightning conductors with their points upon the chimney, are not subjected to the preliminary inductive effect and strain, which may be dispelled by a silent brush discharge.

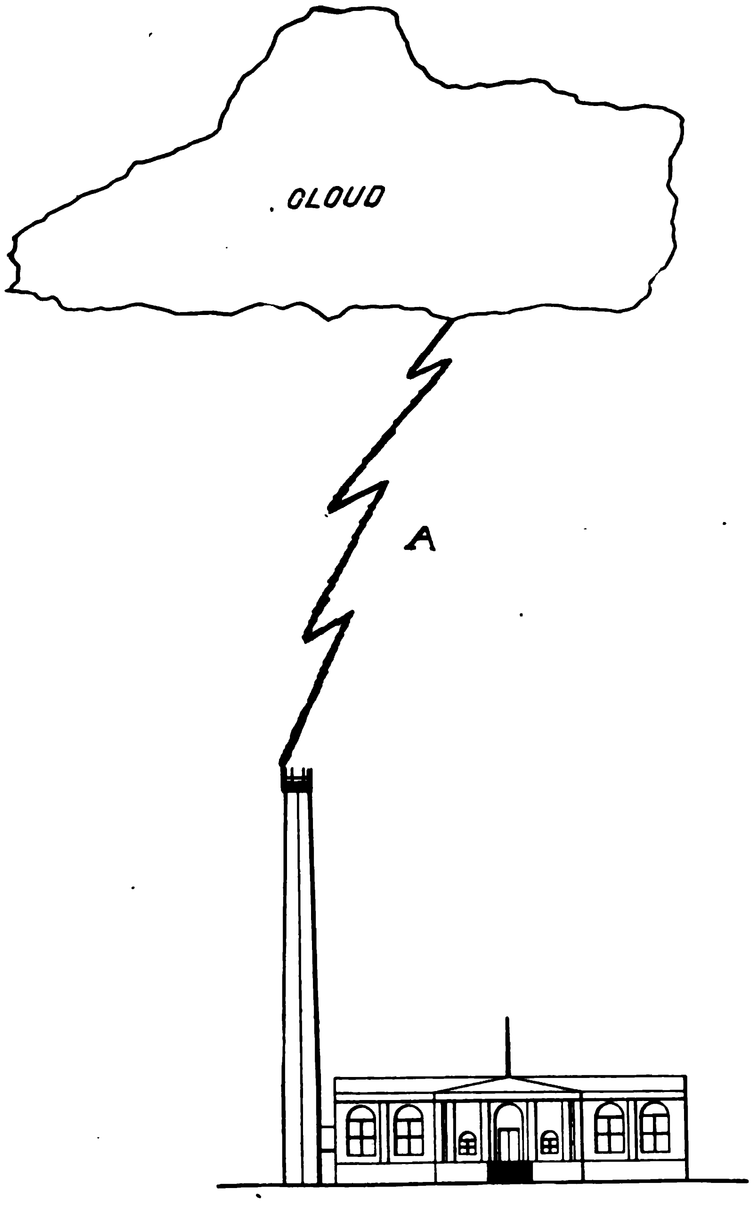


FIG. 1.

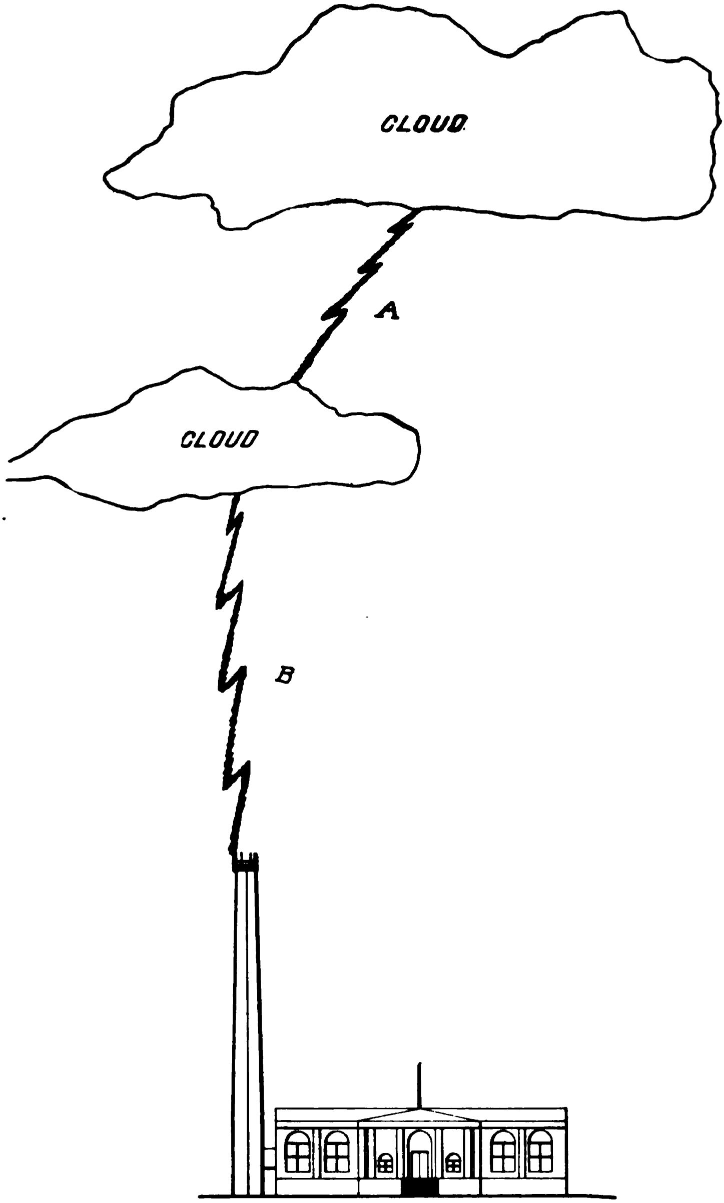


FIG. 2.

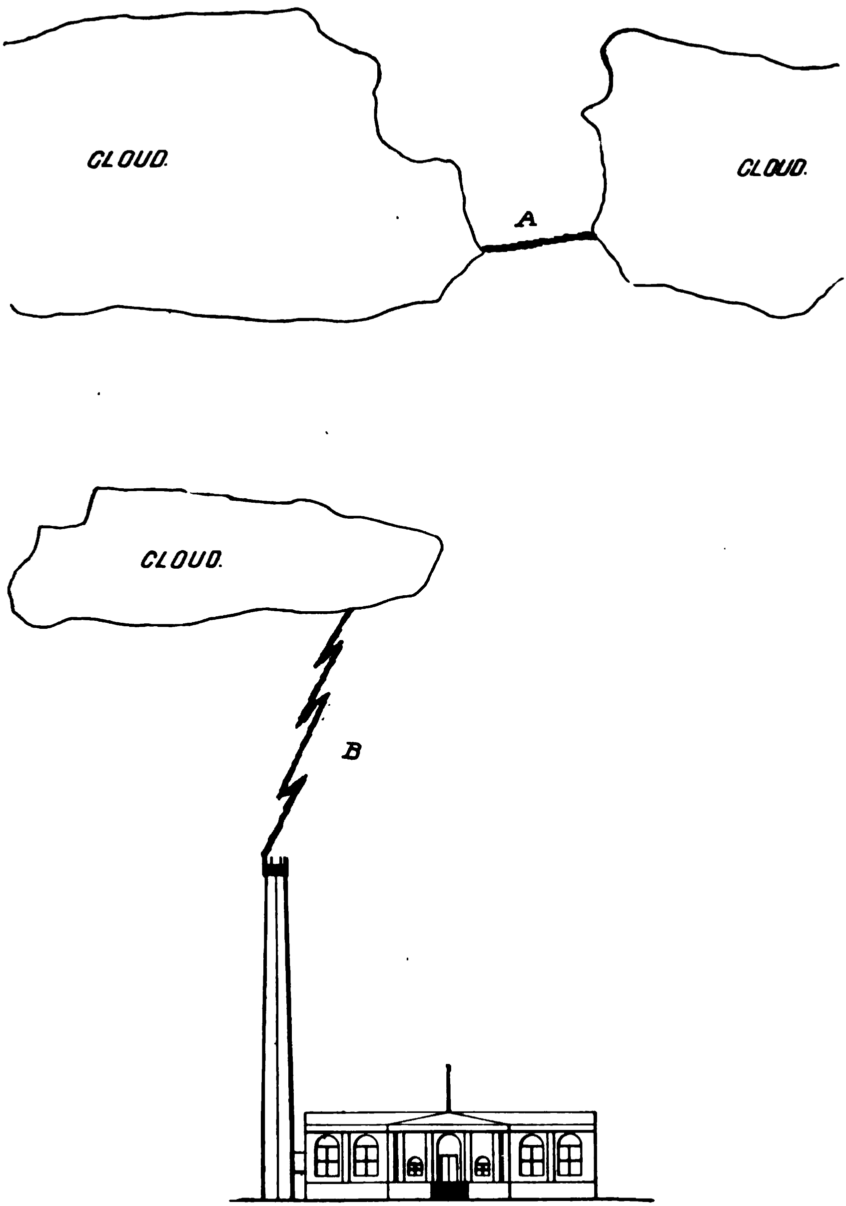


FIG. 3.

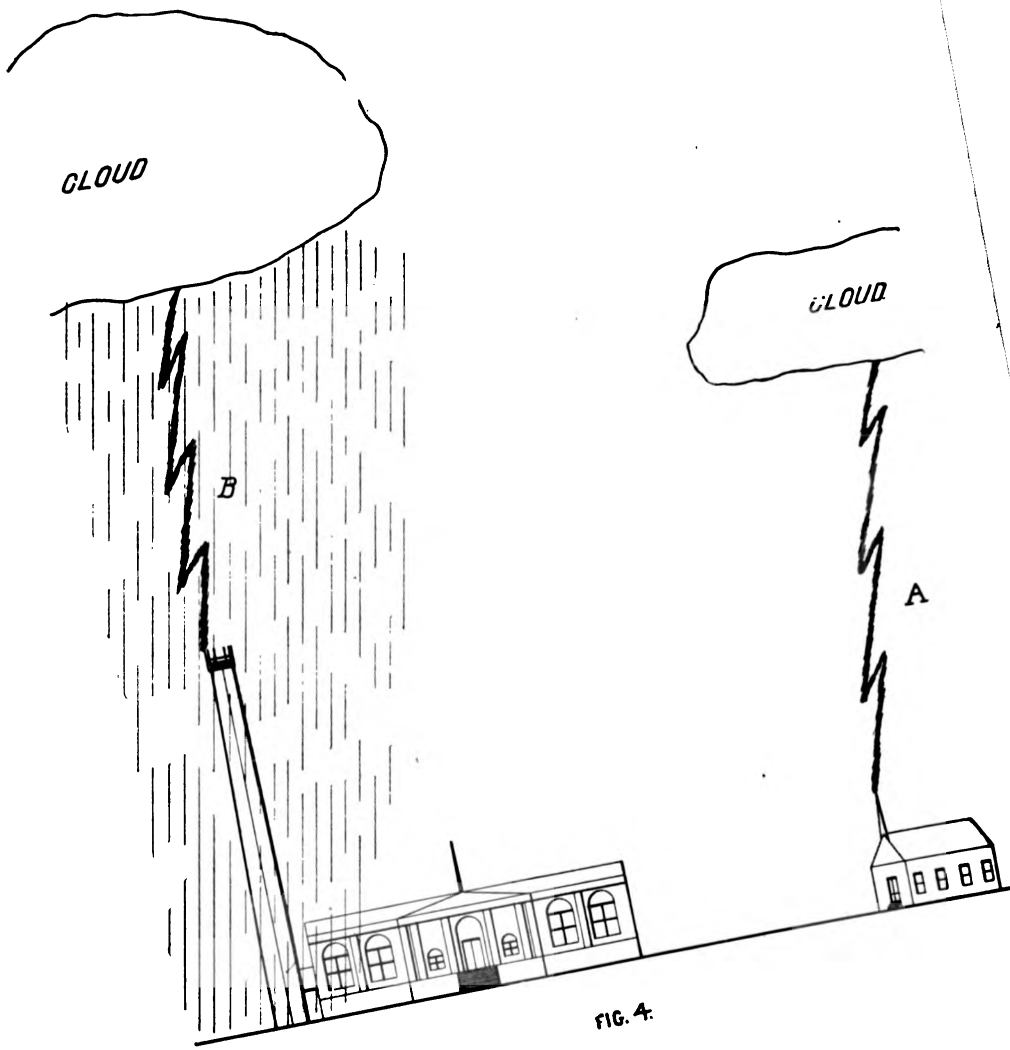


FIG. 4.

In the simple A class of discharge there is a steady, increasing electrical pressure between the cloud and the sharp point of the lightning conductors, allowing them to silently prevent a stroke.

The brush discharge, which is accompanied by the production of ozone, seems to establish a path for the discharge between the cloud and chimney, partly neutralizing its force by so doing. As already pointed out, the heated gases from a chimney act in a similar manner and will lead a discharge to a fireplace, a stove or other source of heated gases, in a dangerous manner. A properly designed and applied rod will in the great majority of cases prevent this discharge down flues and chimneys. In the case of the B class of discharge, this neutralizing action is almost wholly absent, the strain establishing itself so suddenly that an actual lightning stroke takes place at once, the lightning striking with great violence any prominent part of the building. Figures 3 and 4 show further how the A class of discharge may precipitate the B class, which takes place with such sudden violence. The only known means to protect a building against this B class of sudden disruptive discharge is by enclosing it within a wire cage, as shown by Clerk Maxwell.* This cage-like protection, although impractical to carry out fully for various reasons, has been approached in the present four-conductor rod around a chimney in the following specifications and drawings.

The present writer has demonstrated by experiment that four concentrically placed rods afford a very efficient cage-like protection to chimneys constructed on a miniature scale and subjected to the high-potential, high-frequency electrical discharge from a Tesla oscillator capable of striking through an air gap of four feet.

The experimental equipment for studying the behavior and effects of high-potential, high-frequency discharges upon a model chimney is illustrated in Figure 5. It is estimated that the model chimney and conductors are subjected to an electrical discharge at a pressure of about 1,800,000 volts, with

* "Philosophical Magazine," August, 1899.

a frequency of 200,000 oscillations per second. As will be seen, means are here provided for the establishment of an ascending current of heated gases within the chimney, by means of a Bunsen burner. It will be noted that means are also provided for breaking or opening the conductor, and also for introducing impedance in the circuit to earth, which in the present case is the other terminal of the secondary of the high-

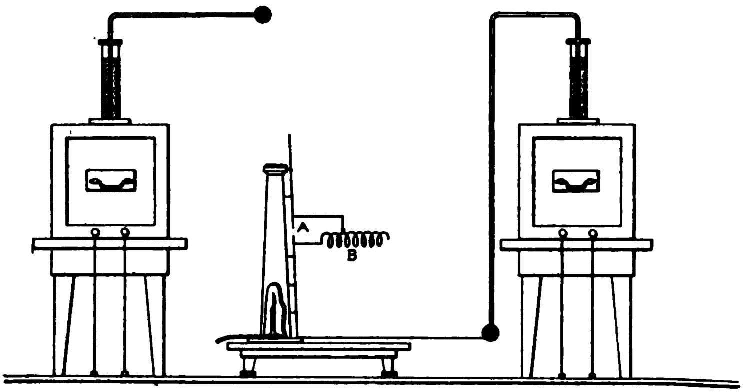


FIG. 5.

potential air core transformer designed for the very high electrical pressures and high frequencies. By lengthening the gap in the conductor at A and increasing the impedance at B, the conductivity of the conductors to these high-potential,

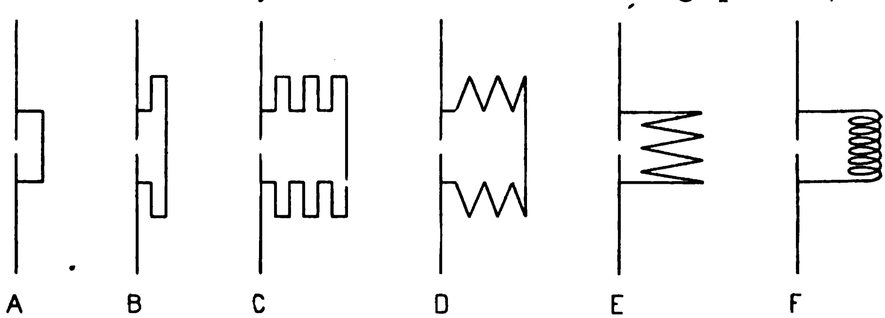


FIG 6.

high-frequency discharges can be balanced against the conductivity of the heated column of gases within the chimney, which consist principally of carbon dioxide and water in the

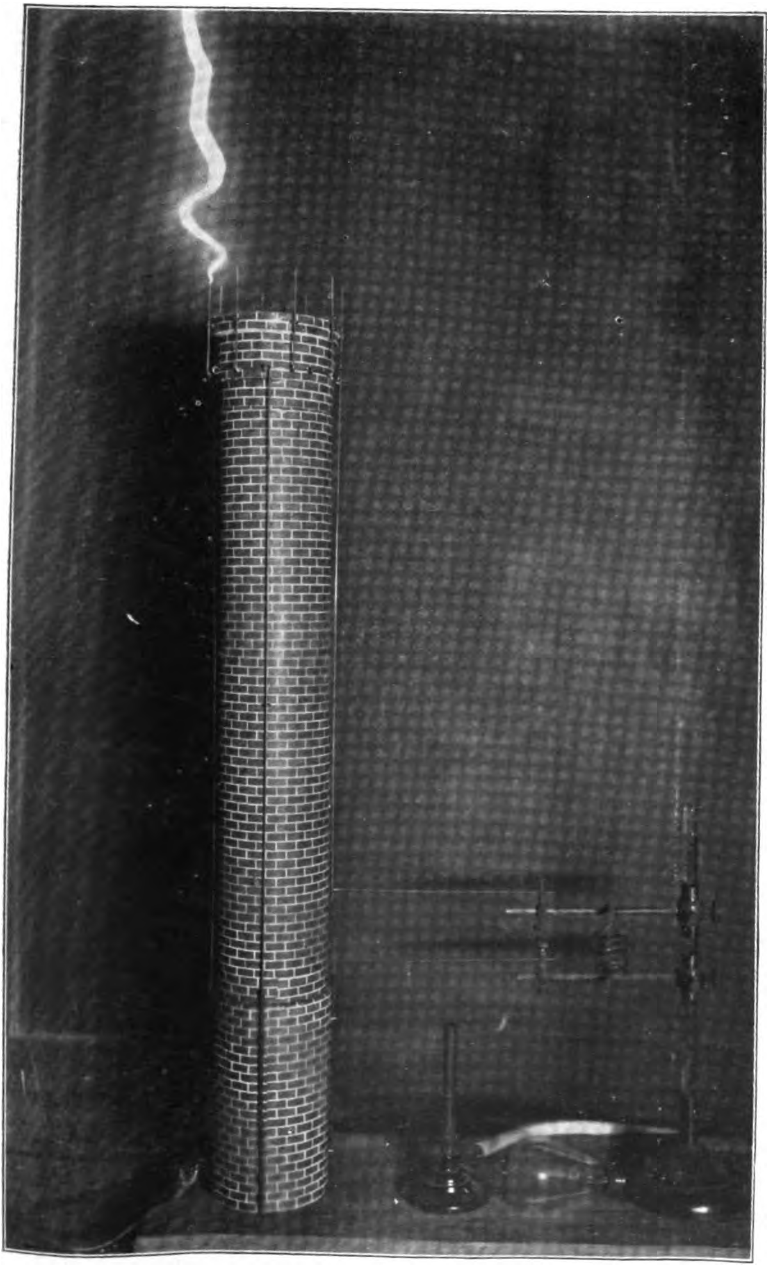


FIG. 7.

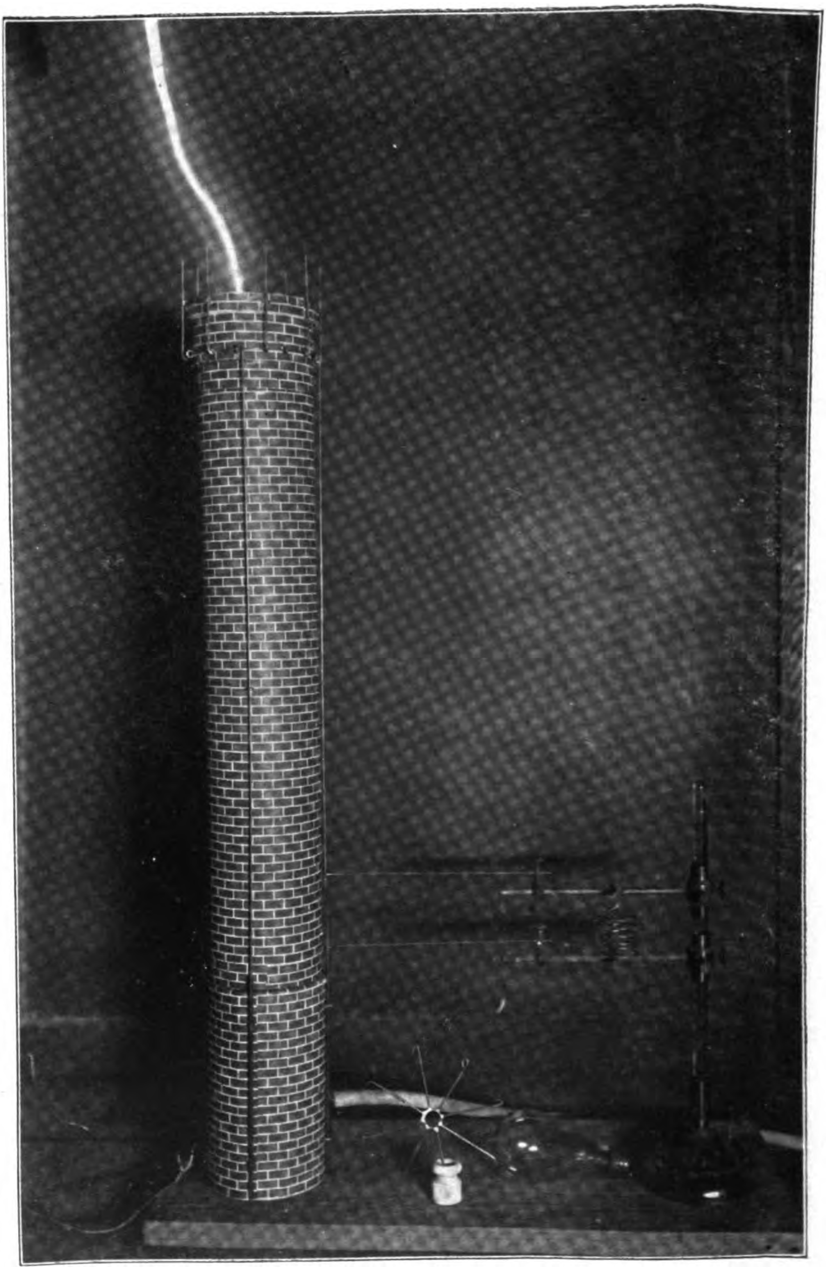


Fig. 8.

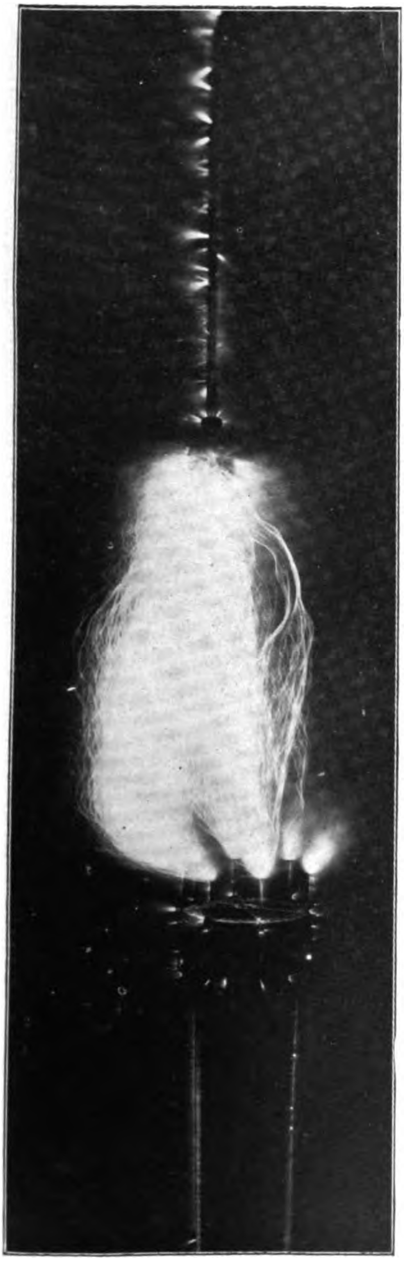


Fig. 9—A.

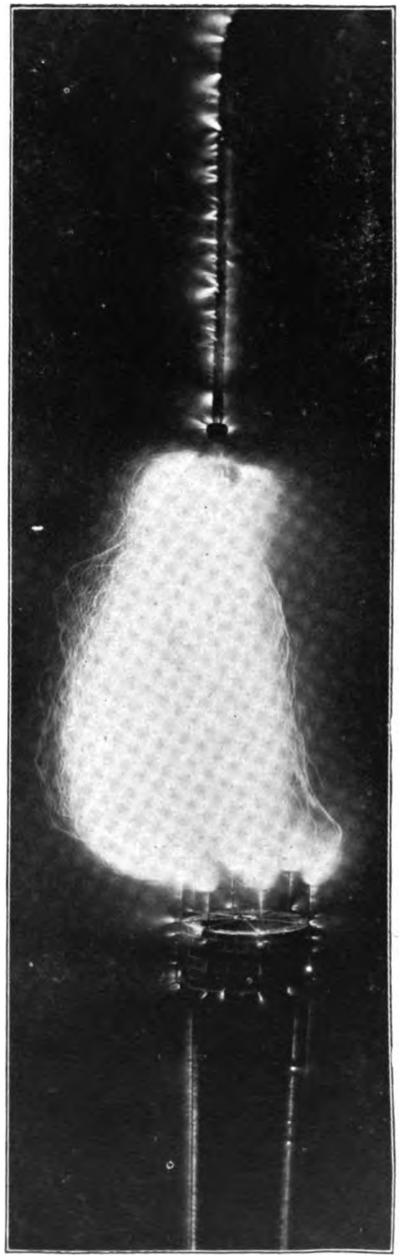


Fig. 9—B.

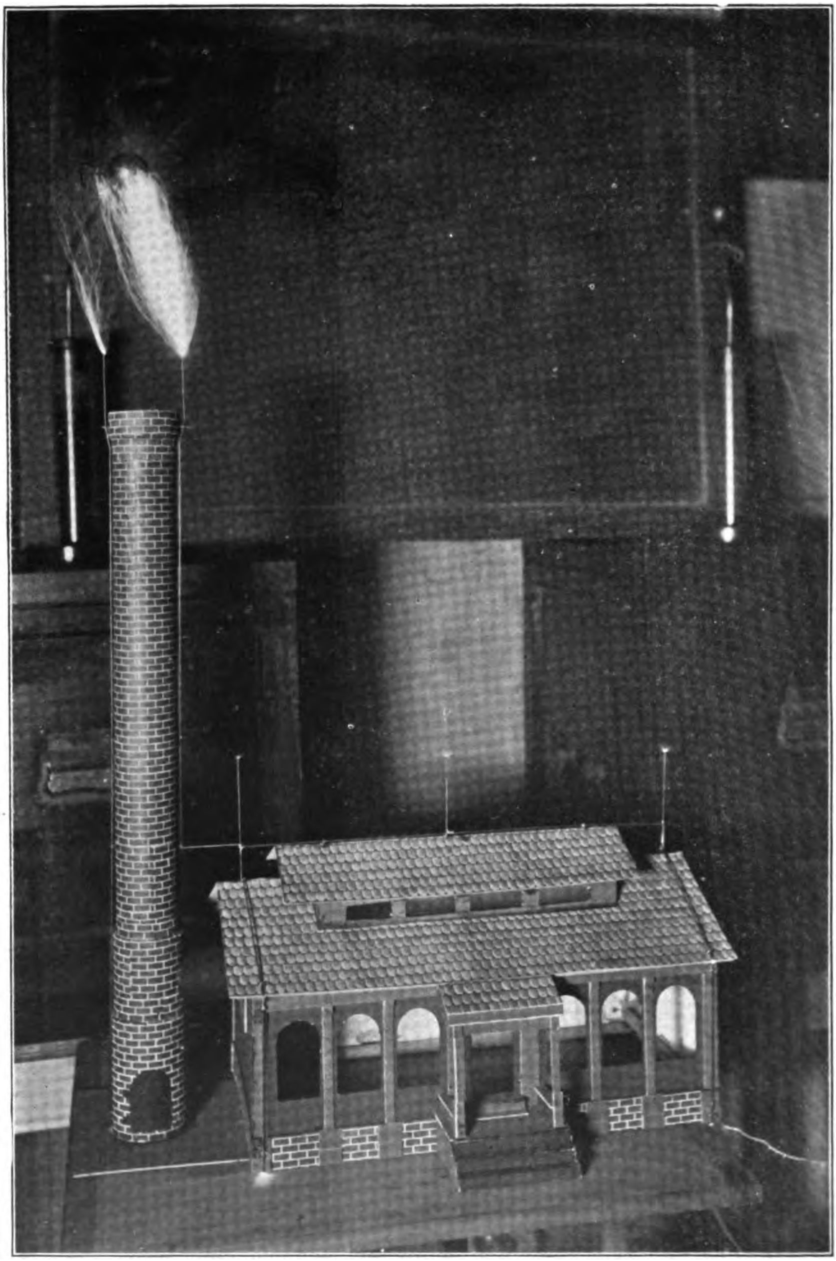


Fig. 10.

form of vapor resulting from the combustion of the gaseous fuel within the Bunsen burner.

It is easily shown that because of the great "skin effect" of high potential discharges of high frequency a discharge may descend within the chimney, tearing it to pieces from the inside, in spite of the conductors on the outside. To protect the chimneys from destruction from this character of stroke a heavy copper spider is provided as indicated in the drawings accompanying the following specifications. This spider covers the top of the chimney, the radial limbs of which run to the vertical rods. A discharge may now be directed to the ascending current of hot gases, but will strike the spider and be dissipated radially at right angles to the conductors running to earth. As it has been held for many years that lightning strokes will not readily change their directions abruptly, or pass around sharp corners, it was deemed desirable to carry

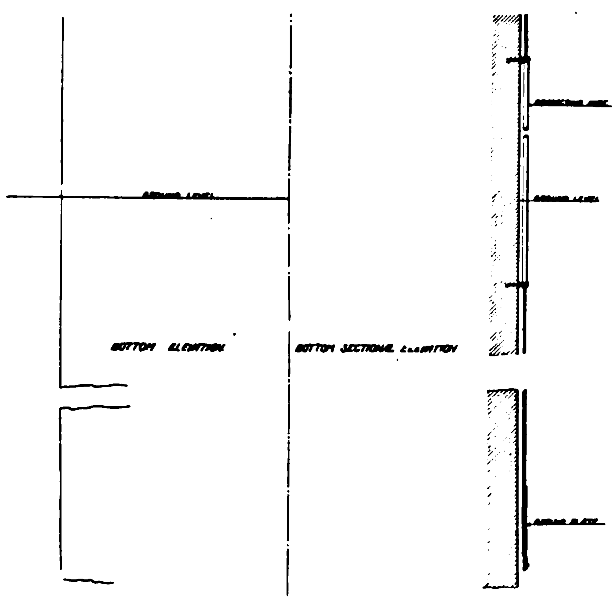
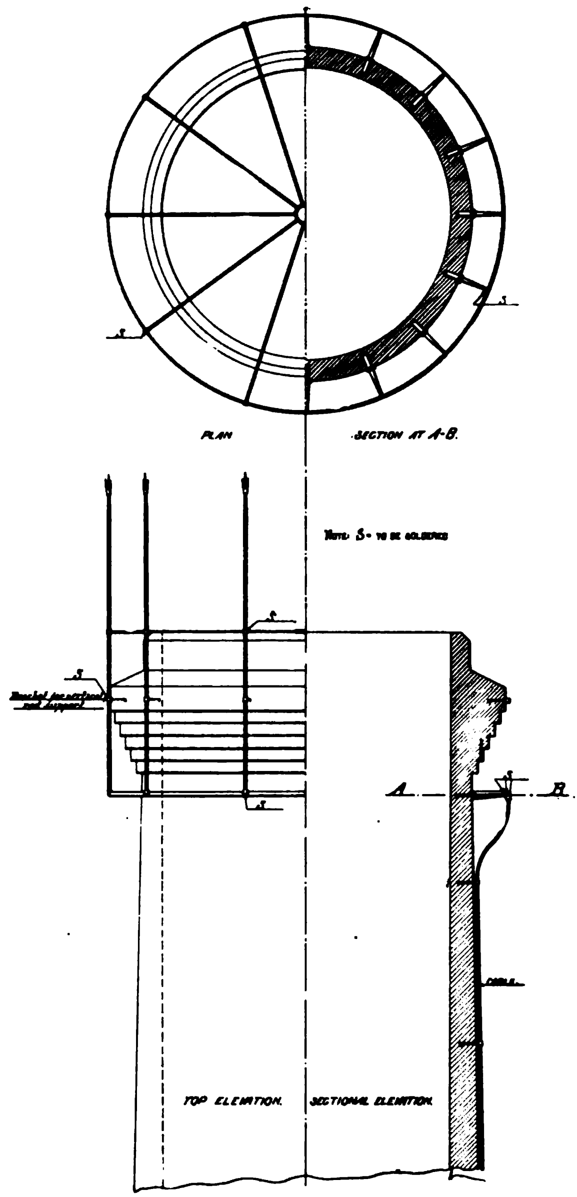


FIG. 10—A.

GENERAL ARRANGEMENT OF CHIMNEY BASE.



GENERAL ARRANGEMENT
OF CHIMNEY TOP.

FIG. 11.

on some experiments to determine whether or not it was safe to eliminate from the design all easy or rounded bends.

As the installation of the spider referred to required for the protection of the interior of the chimney a right-angle deflection to the conductors to earth, experiments were made with conductors of various shapes involving sharp bends and reversals of direction. Figure 6 shows some of the forms of conductors experimented with arranged across a variable air gap. It was demonstrated that these high-frequency discharges of enormous voltages pass readily around various kinds of sharp bends and reversals in direction, as shown at a, b, c, d, e, in the figures. With any path of the nature of a spiral as shown at f, however, we of course introduce high impedance, as naturally would be expected with discharges of this kind. A full description of all experiments made, or a complete record from the writer's note book upon experimental lines would hardly be in place here, for the present

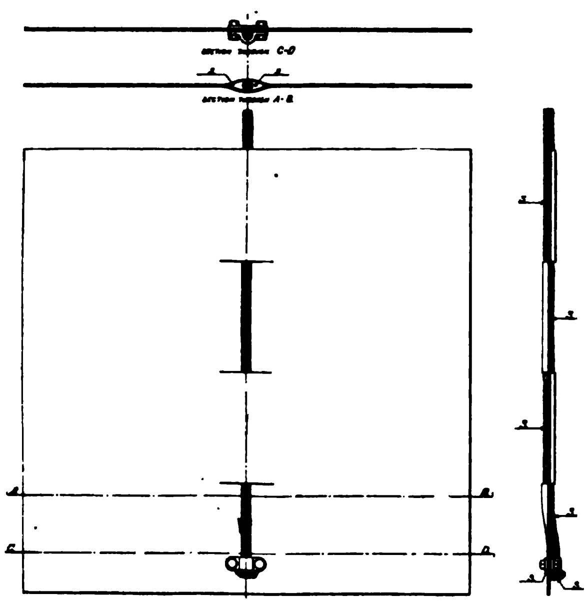
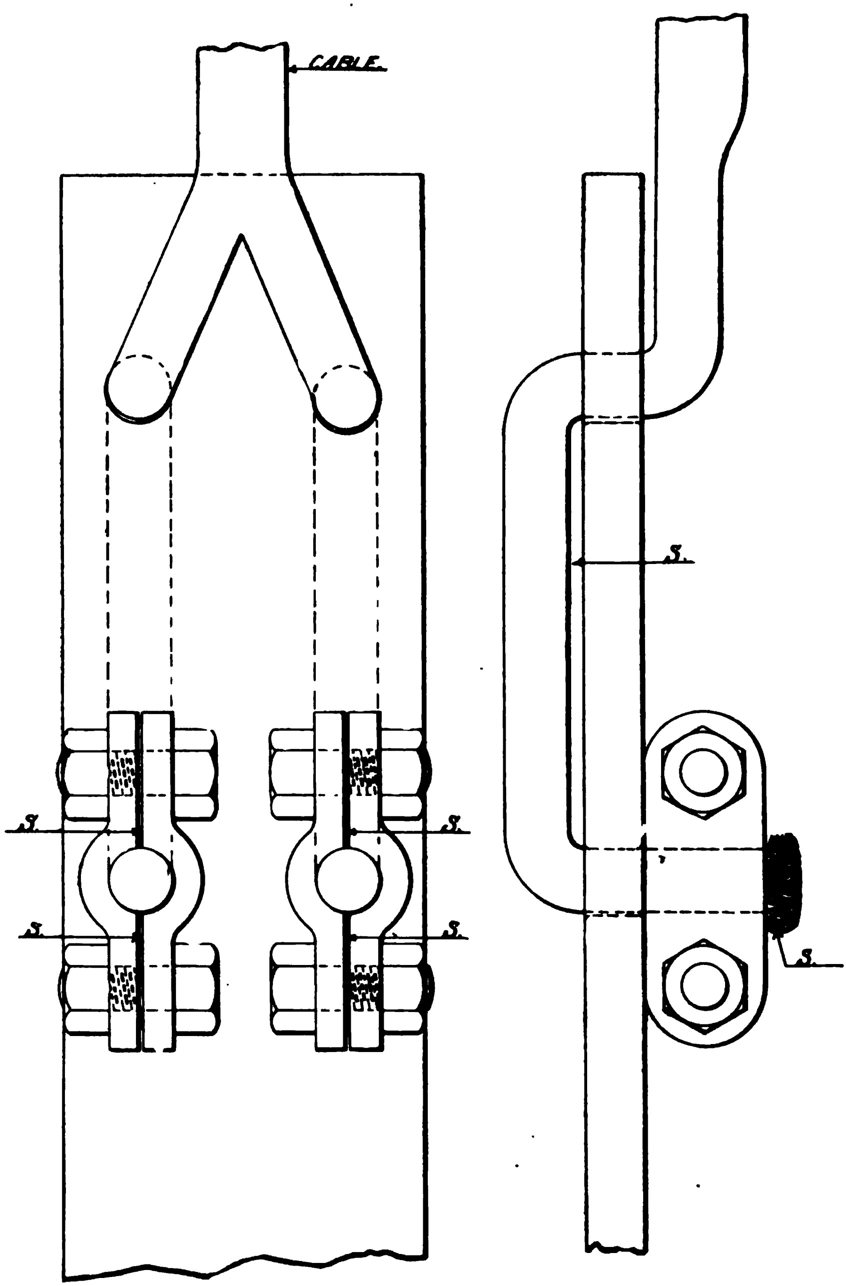


FIG. 12.

GROUND PLATE, TYPE NO. 1.

NOTE.—S = to be soldered.



Note: S = To be soldered.

Fig. 13.

Ground Plate, Type No. 2.

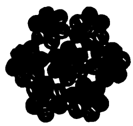


FIG. 14

SECTION OF CABLE.

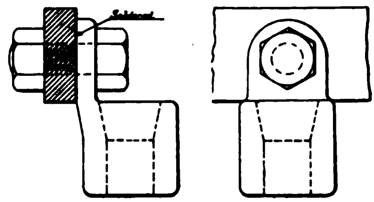


FIG. 15.

DETAIL OF CONDUCTOR ATTACHMENT TO COPPER RING

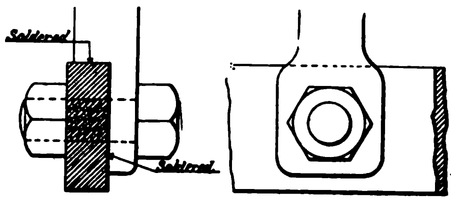


FIG. 16.

CONNECTION OF MULTIPLE TIPPED RODS AND COPPER RING.

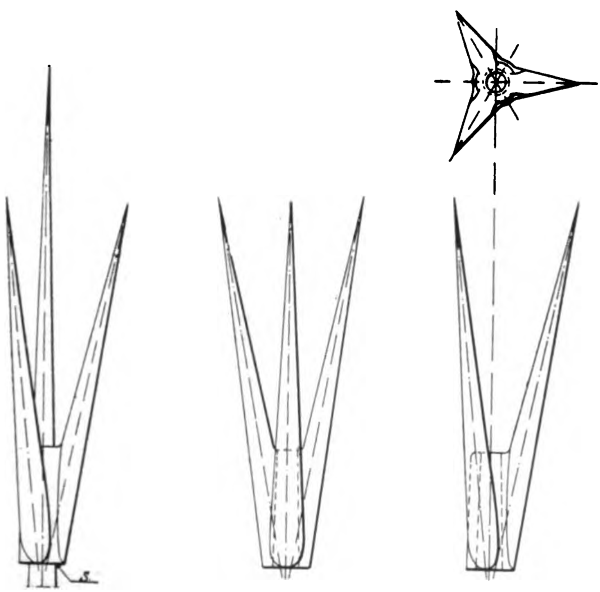


FIG. 17.

CAST POINTS.

NOTE —S = to be soldered.

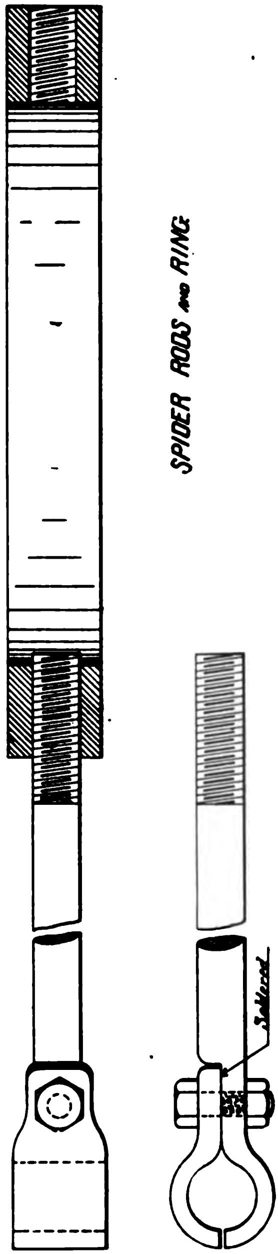


Fig. 18.

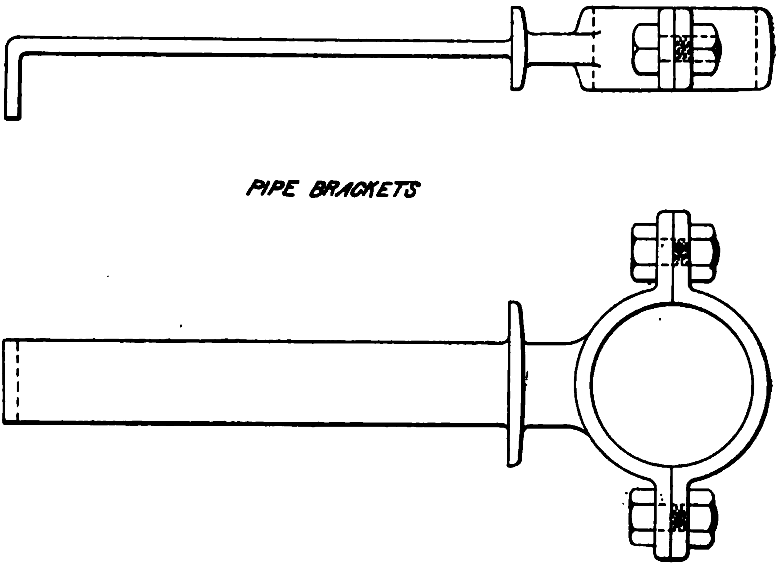


FIG. 19.

article endeavors rather to present the engineering features dealing with the practical application of conductors to tall brick chimneys. It has been the intention, therefore, to introduce only such general experiments as to show reasons for the salient features of the general design of the specifications which are given at the close of this paper. Figure 7 is from

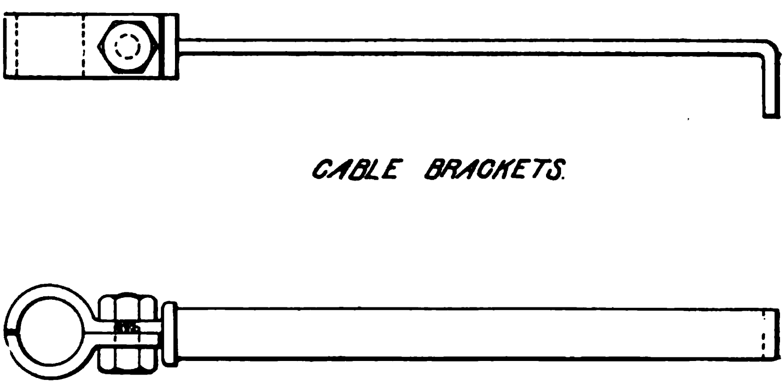


FIG. 20.

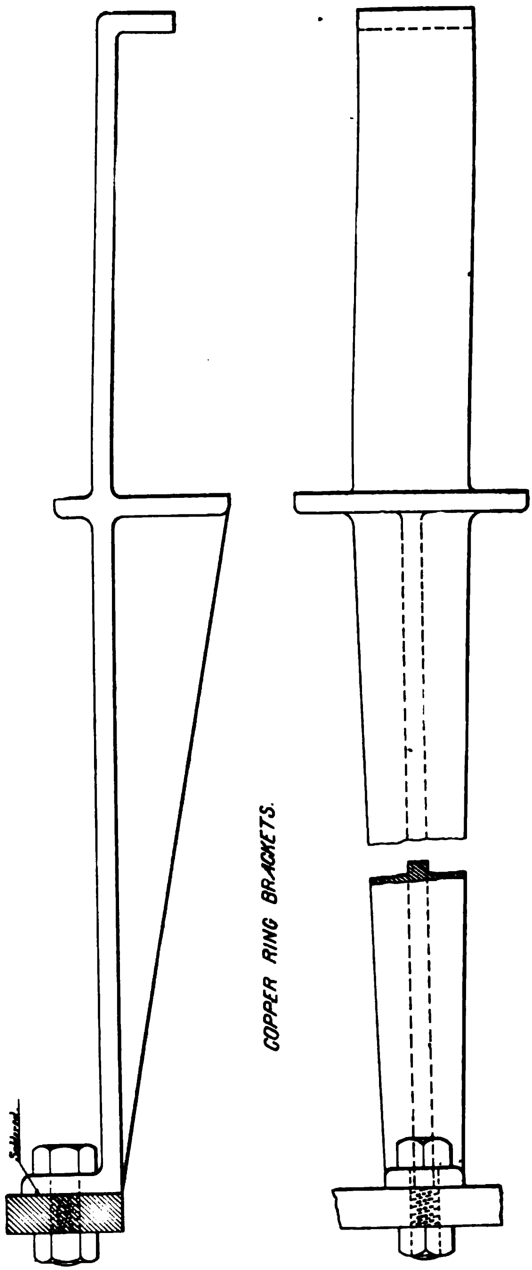


FIG. 21.

a photograph of the model pasteboard chimney experimented with. This illustrates how a B class of discharge may strike a sudden and vicious blow of oscillatory character. Here we must have ample copper area as well as conductor capacity to carry the discharge current to earth.

In this experiment the chimney top is provided with a copper spider spanning the opening, the limbs of which run radially to the four conductors to earth. The scale of the model chimney is indicated by the Bunsen burner at the right in the foreground, as well as by the ordinary 16-c.p. incandescent lamp lying alongside. The stand with glass column at the extreme right supports the impedance coil, which may be varied in inductive value and be inserted in any of the conductors to ground under experiment. In the photograph, Figure 8, we have the same chimney represented, but with the copper spider removed and exhibited at the base, the Bunsen burner being lighted within the chimney. The flash is shown striking down the interior of the chimney conducted by the carbonic acid gas and water vapor from the burning fuel in the burner. Figure 9 shows the same chimney subjected to the discharge from an electrical oscillator at close range, where the electrical strain is so great that all the eight points about the chimney top take part in dissipating the energy. This experiment represents conditions analagous to the class A discharge. Figure 10 shows a model pasteboard power house and chimney with system of conductors under the discharge of an oscillator working at diminished power but at closer range. Brush discharges are quite plainly visible from the tips of the vertical rods on the roof top. The importance of a sufficient number of sharp points surmounting all conductors cannot be overestimated.

The following is translated from the German upon the subject of providing ample streaming or equalizing points. *"The buildings of a manufacturing plant are connected by means of barbed wire strung on poles. By the connection of the barbed wire to the walls of the deeper-lying buildings, a Farraday-

* "Ueber den Blitzschutz auf Sprengstoff-Fabriken insbesondere Nitroglycerin-resp. Dynamit fabriken. Direktor Knight, Dynamitfabrik, Krümmel."

cage-like protection is secured on the one hand, and on the other hand, on account of the numerous points on the wires, a more intense equalization of the electric tension between the earth and the clouds is reached. In a dynamite factory, for example, without cartridge buildings, it is necessary to have a combination of 7 buildings with about 6,000 meters of barbed wire, 1,200,000 barbed points and 70 pole earth-conductors, and it can be readily understood that by this arrangement a stroke of lightning on such a system of barbed-wire nets dissipates its electric energy and is led to earth without leaping aside or forming secondary currents. Assembled nitroglycerine factories, dynamite magazines, and in part also the drying houses, possess this barbed-wire-netting protection, so that the aggregate of barbed wire points on the floor space of the Krümmel factory consists of more than 5,000,000."

For the complete protection of a central power plant, therefore, its roof and trusses, together with all other masses of metal without and within the building should be metallically connected with chimney conductors as well as to light rods running along the top of all roofing and other prominent parts of the building. Sharp points should be placed at close intervals somewhat analogous to the protection afforded by the barbed-wire lines as described in the German article. As the architecture of the building must necessarily dictate the precise arrangement of conductors, the specifications as herein given pertain only to the protection of the chimney which, if properly provided for, because of its towering height, affords also good protection for the building. The present paper closes with the following specifications as approved for the protection of the brick power-plant chimneys of the Navy.

STANDARD SPECIFICATION FOR LIGHTNING RODS.

Chimney Protection for Power Plants.—Lightning conductors shall be laid up in the form of a seven-strand cable and each strand laid up with seven copper wires of No. 10 B. & S. gauge. For chimneys of 50 feet and less in height, two lightning conductors shall be used. For chimneys over 50 feet up

to and including 100 feet, three conductors shall be installed. For chimneys higher than 100 feet, four conductors shall be nstalled. All heights to be considered from ground level. All conductors or cables shall be symmetrically arranged about the chimney with one cable on the prevailing-weather side of the chimney. Said lightning conductors or cables to be securely attached both mechanically and electrically to independent pure copper earth plates or bars. In cases where the chimney foundations have already been filled in, instead of earth plates earth terminals may be used, composed of pure copper bars 3 inches by ½ inch by 3 feet. In all cases the lightning-conductor terminals shall extend to the ground water level, and in no case shall they extend to less than 15 feet from the ground surface. Earth plates shall consist of pure copper 3 feet by 3 feet by ⅛ inch.

Application of Conductors to Chimney.—Each lightning conductor shall be secured to the exterior of the chimney by means of bronze or brass anchors, without the intervention of any insulators or insulating material whatever. The brackets for attaching ring or conductors to chimneys to be of high-grade bronze or brass, composition of same to be submitted for approval, and to be fitted with approved clamps for securely gripping said conductors and making a good electrical connection therewith. The tongues or shanks of the anchors or brackets shall enter the masonry of the chimney a distance of at least 6 inches, and shall be at least ⅛ inch in thickness by 1 inch wide, terminating in a suitable head or angle, to prevent the anchor from being pulled out of the masonry. Anchors to be attached to conductors at intervals of not over 10 feet, and sweated to the conductors with solder at intervals of 50 feet. Conductors to terminate within 5 feet of the top of the chimney, and to be connected through the agency of suitable brass or bronze fitting and be soldered to a 1½-inch by ½-inch ring of copper attached to the periphery of the chimney by brackets spaced not over 2 feet apart; said brackets to enter the brickwork a distance of at least six inches and to be of approved design, with a tongue at least 1½ inches in width and

¼ inch in thickness, with a suitable angle or head to prevent pulling out. All joints in the continuity of said copper ring as well as between the continuity of the ring and conductor or conductors running down to the ground bars or plates, and including the latter, to be scraped bright and, after making a secure mechanical joint, to be "sweated with solder." Said solder shall consist of ½ lead and ½ tin. All joints when finished shall be thoroughly washed off with water to remove every trace of soldering salts, acids or other compounds used. All joints secured by bolts or screws to be locknutted. In applying conductors where the chimney is already constructed holes shall be drilled in the brickwork and said anchor brackets and anchors grouted in, the best Portland cement being used.

Terminal Rods for Lightning Conductors.—Copper ring shall be connected through the agency of clamps, insuring a good mechanical and electrical joint, with vertically-arranged copper rods, at least ¾ inch diameter and 10 feet in length. The joints to be "sweated with solder" as before described. Copper rods to be placed equidistant around this ring, and supported in a rigid position vertically through the agency of additional anchors set in the masonry and a copper spider resting on chimney top as shown in the drawings. Rods to be arranged with a uniform spacing of practically 4 feet. This is taken to mean, for example, that ten such vertical rods shall be provided for a chimney of 12 feet outside diameter of chimney at top.

Discharge Points.—Each rod shall terminate in a 2-point aigrette, each spur or point of this aigrette to be at least 3¾ inches long, the bases of which spurs shall be at least ⅝ inch in diameter, tapering to a sharp and well finished point; said aigrette to be provided with approved means to secure a strong mechanical and electrical joint with the vertical rods heretofore described and to which it is attached. The joints shall be "sweated with solder" as heretofore described.

Chimney-Base Protection.—All lightning conductors shall be enclosed at bottom with a heavy galvanized iron pipe of 1½ inches diameter, and extending 3 feet into the soil and 10

feet above. Said iron pipe to be provided with approved brackets to securely hold it to the chimney; brackets to be not over 3 feet apart.

Inspection.—No scaffolding shall be removed before inspection of the lightning-conductor installation shall have been made by a Government representative.

Note.—This specification is to be accompanied by a drawing, furnished by the Bureau of Yards and Docks and approved by the same, giving detail parts.

TORSIOMETERS.

TORSIOMETERS AS APPLIED TO THE MEASUREMENT OF POWER IN TURBINES AND RECIPROCATING ENGINES.*

By Archibald Denny.

When the suitability of the turbine method of propulsion for commercial work was proved by the success of the *King Edward*, built by my firm in 1901, it became apparent to us that it would be highly desirable to have a method of ascertaining the horsepower transmitted by the turbine shafts to the propellers. Until that problem was solved we could only work from the boiler to the propeller, and the efficiency of the turbine and the propeller must be lumped together. It is not possible to "indicate" the turbine in the same way as is done for a piston engine, although I may say that a fair approximation can be got by ascertaining the fall of pressure through successive expansions by means of pressure gauges fixed to the turbine casing.

Some fifteen years ago we had made numerous experiments with factory shafting, endeavoring to ascertain the absolute torsion of a shaft while running, and it therefore immediately occurred to me that this was the proper direction in which to attack the problem. We had tried various methods, principally using pierced discs and beams of light, but with very partial success. We had not tried any method involving the use of electricity, and I therefore arranged for experiments to be made by this method on one of our factory shafts. The first trials were made by fixing discs on the shaft at a considerable distance apart, so as to get a reasonable amount of torque. The discs were of insulating material, and each had

* Paper read before the Institution of Naval Architects, March 21, 1907.

a contact point arranged at its periphery in such a manner that the point made momentary contact with a metal tongue or brush once in every revolution of the shaft. The contact points were connected to the shaft, and the metal brushes to a battery and a telephone receiver. The method adopted was to first adjust the brushes so that both made contact with the points simultaneously when the shaft was revolving, but transmitting no power. When transmitting power the shaft was, of course, subject to a certain amount of torsion, and thus the brushes were put out of simultaneous contact. One of the brushes was then moved round its disc concentrically, until simultaneous contact was once more established. The amount of this shift gave a measure of the torque on the shaft, and to ascertain the correct amount of this shift the telephone receiver was placed to the ear, no sound being heard except when both brushes were in contact with the respsctive contact points, when a loud "tick" was heard. The principle was thus of extreme simplicity, and the method of carrying it out seemed at first equally simple; indeed, I may say that this first rough apparatus, which was quite successful, cost only a few shillings to make. We then set about making more accurate and elaborate apparatus on the same lines, to be fitted to the *Queen Alexandra*, which was nearly ready for trial, with an assured hope of getting satisfactory results.

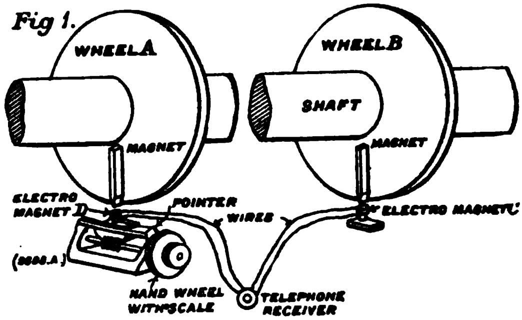
The factory shaft on which we made the original experiments ran about 120 revolutions per minute, but the revolutions of the *Queen Alexandra's* side shafts were over 700, and when we came to make experiments at this high speed we found the new apparatus was useless, as no certain sound could be got. We tried many forms of contacts, and after numerous experiments we did succeed in the *Queen Alexandra*, with revolutions about 750, in getting some fairly consistent results; but it was impossible to be quite certain of the exact point at which the make-and-break in the circuit took place, and we were never quite sure of our results; still we had made a great step in advance.

Mr. Charles Johnson, a member of our staff, who assisted

in working out this problem, and was closely connected with it from the first, thoroughly appreciated the difficulties, and realized the desirability of getting away from the unreliable rubbing contact, and he ultimately succeeded in solving the problem in a most ingenious way.

Fig. 1 and Fig. 8 show his original solution. Two gun-metal wheels, A and B, were fastened to the shaft at a definite and known distance apart, the distance being as great as possible. On each wheel a permanent magnet, with a sharp, chisel-shaped edge, was fixed radially at the periphery of the wheel and with the sharp edge parallel to the shaft. At one end a soft-iron electro magnet C, wound with fine wire, similarly chisel-shaped, was fixed, so that the moving magnet

Fig 1.



passed directly over the electro magnet once in each revolution. At the other end a similar electro magnet D was mounted on a screwed sector, and wires from these electro magnets were led to a differentially-wound telephone receiver. If the shaft revolved without transmitting power, the permanent magnets passed the electro magnets simultaneously, and currents of electricity generated in each coil passed through the telephone receiver; but the currents being equal and opposite, no sound was heard. When the shaft transmitted power the permanent magnets passed the electro magnets at different times, and hence a sound was heard in the receiver. By turning the hand wheel shown in the diagram a new position of silence could be obtained, when it was evident that the two permanent magnets were again passing the electro mag-

nets simultaneously, and the amount of torque could be ascertained from the reading of the sector screw.

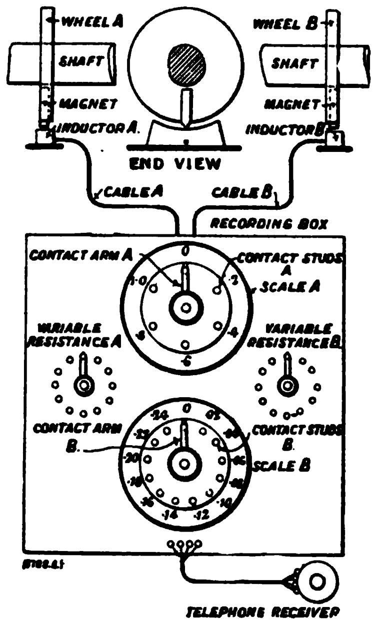
This instrument was used with great success on a number of the turbine vessels built by my firm, but it required the operator to work in the tunnel, where, even with turbines, there is some noise and disturbance; and to overcome this objection Mr. Johnson designed another arrangement, which is illustrated in Fig 2. The gun-metal wheels and permanent magnets remain as before, but in place of the chisel-shaped electro magnets, soft-iron sector cores, marked inductors A and B, take their place. These cores are wound with a series of coils of very thin wire, so fine, indeed, that each coil, with its dividing wall in one of the sectors, only occupies a space of 0.02 inch. This sector, or indicator, as we term it, is fitted underneath one of the gun-metal wheels, and a similar one is fitted at the other wheel, but in this case the coils are further apart—namely, 0.2 inch; the reason for this will be afterwards explained.

In Fig. 2 is also shown the recording box, which will be seen to consist of two circles fitted with contact studs and movable contact arms; the upper one, marked "Scale A," is connected with the inductor at the wheel next the propeller, and has six contacts; that is to say, there are six coils, spaced 0.2 inch apart. The other one—"Scale B"—has thirteen contact studs, which are similarly connected to thirteen coils on the inductor at the turbine end of the shaft; these coils are spaced 0.02 inch apart. On the recording box will be seen two side coils, marked respectively "Resistance A" and "Resistance B." These are resistance coils for throwing into series with the differential windings of the telephone receiver, which is connected to the two inductor circuits, which circuits must be accurately balanced before absolute silence can be got in the receiver.

Fig. 2 shows the arrangement of the apparatus for one shaft. With this arrangement it is possible to locate the observer in any part of the ship, cables being led from the inductors in the tunnel to the recording box, and we usually

select some quiet cabin where the observer will be undisturbed. The sound in the receiver at usual revolutions is so distinct that even an untrained observer, after a very few min-

Fig. 2.



utes' practice, can get perfectly accurate results, and powers transmitted can be ascertained from moment to moment.

It is essential, of course, that the correct zero position, from which to start, should be found. In order to do this, zero lines, or, rather, grooves, are cut into the cover plate of the inductors A and B, directly over the zero coils; and in setting up the instrument the chisel-shaped magnets are brought directly over these lines, by first dropping them exactly into the grooves, and then raising and fixing them to just clear the top of the inductors. But we do not depend upon this method alone, because any inaccuracy in setting up the inductors might bring in small errors. We therefore get an absolute zero by running each shaft light alternately; that is to say, before the commencement or after the completion of

the trials, each shaft is alternately run light, the vessel being driven by the other two screws. It is true that by this means we do not eliminate the effect of friction in the turbine and shaft bearings; but this must be very small, as we find our original method of setting the zero, when carefully done, is always confirmed by this method of checking the same.

To take a reading after the instrument is once set, and the resistance of the two telephone circuits is adjusted by means of the variable resistances A and B, all that is necessary is to turn the movable arm on scale B round the various studs until there is silence in the telephone, when the amount of torsion is immediately read off the scale. If no such position be found, it means that the shaft is being twisted more than is covered by this scale; the arm A is then turned to the first contact, and the arm B is again swept round the circuits. If silence be still not obtained, the arm A is turned to the second contact, and so on, the combined range of the scales being altogether 1.24 inches, which is more than sufficient to measure the maximum torque usually obtained. From torsion experiments on the shaft described below, made previous to its being fitted in the ship, or from a formula also given below, the factor by which this reading is to be multiplied is obtained, and the power is got by a simple multiplication sum.

This instrument is made for one, two, three and four shafts; for each shaft a group of circles as shown is required. By means of the contact arms and studs the various shafts are thrown into circuit with their receivers, and readings are taken from three or four shafts in a very short time. This instrument is not suitable for very low revolutions, as the induced current becomes too weak to make a distinct sound in the telephone; but it is suitable down to about 100 revolutions per minute. For lower revolutions the instrument first described is employed, but the operator must then be in the tunnel.

The resting of the shaft to get the torsion scale previous to its being fitted on board is done by fixing rigidly one end of

the length of the shaft to be used for the trials, and at the other end fitting a lever; this is loaded with weights, so as to get the scale of torsion moment. For turbine shafts, which are small in diameter as a rule, this is not a serious operation, nor, indeed, have we found it so even for larger ordinary twin-screw shafts, although, of course, the weights used have to be much greater. We have tested all the shafts for the turbines and other vessels on which we have used these instruments, and we find that the torsion is given very closely by the following formula:

$$\theta^0 = \frac{W R L}{K d^4},$$

where

θ^0 = angle of torsion in degrees.
$W R$ = foot-pounds turning moment.
L = length of shaft in feet.
d = diameter of shaft in inches.
K = coefficient.

K may be taken at 140 for mild-steel shafts where there are no couplings, or, if there be any, by deducting their length

Fig. 3. TWIN SCREW STEAMER SHOWING INDICATED H.P. AND SHAFT H.P.

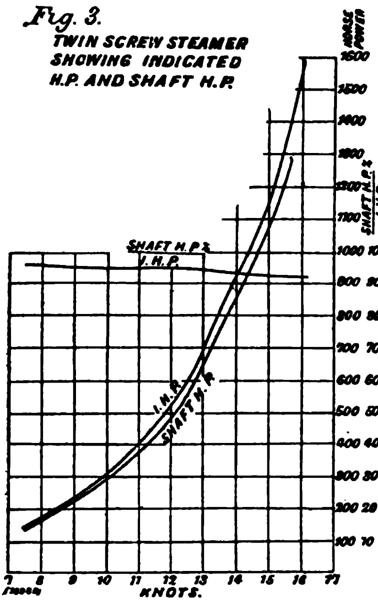


Fig. 4. TWIN SCREW STEAMER SHOWING INDICATED H P AND SHAFT H.P.

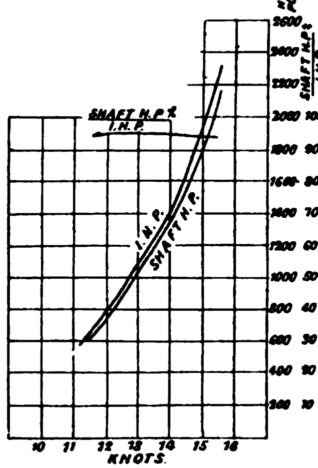
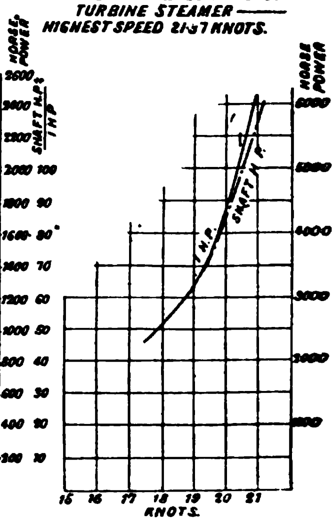


Fig. 5 TWIN SCREW STEAMER —— HIGHEST SPEED 20·7 KNOTS. TURBINE STEAMER —— HIGHEST SPEED 21·37 KNOTS.



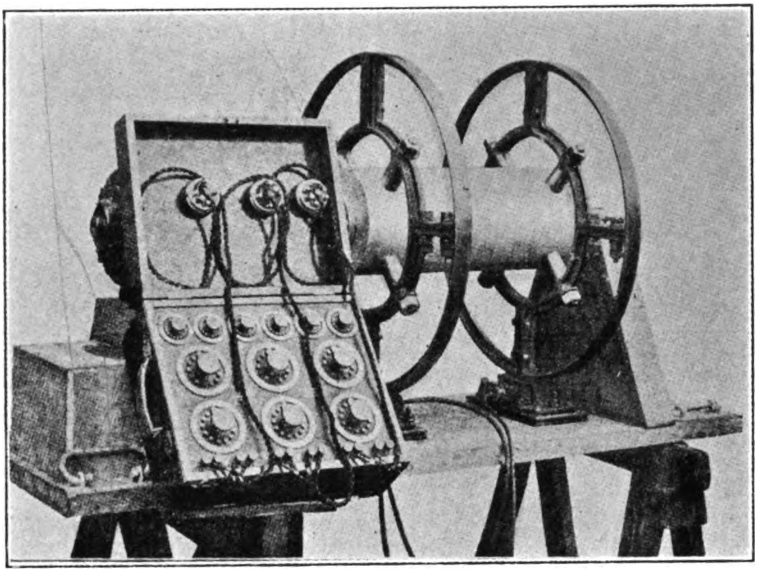


Fig. 6.

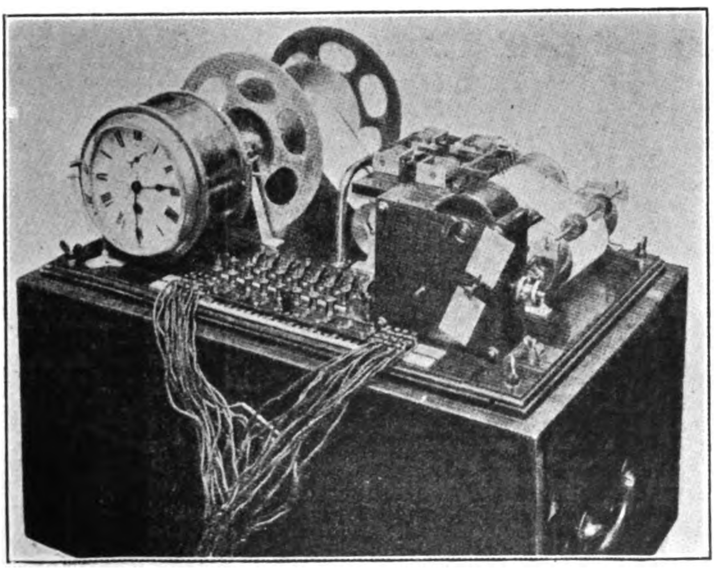


Fig. 7.

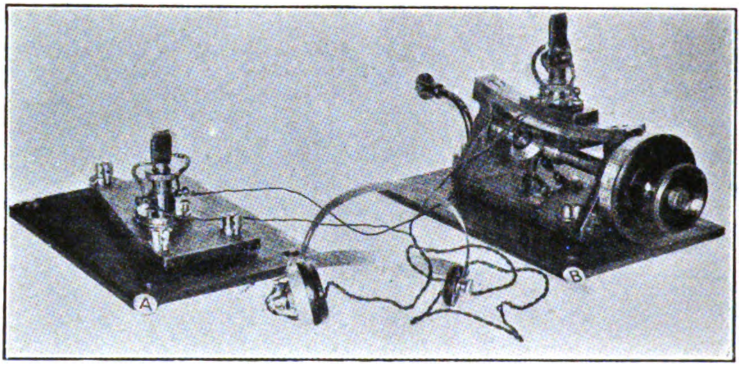


Fig. 8.

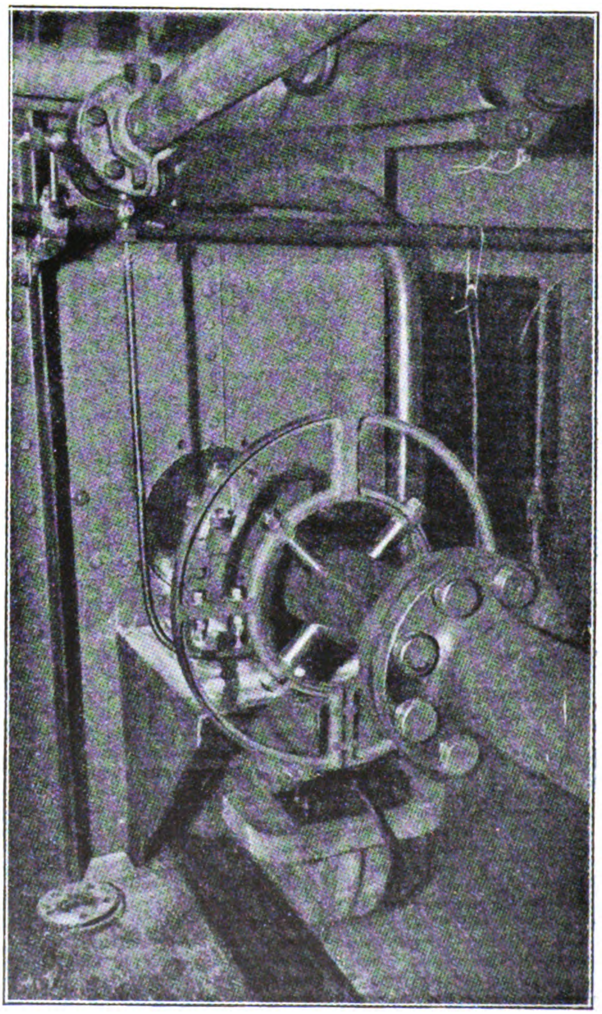


Fig. 9.

from the length of shaft twisted—that is, we assume that the couplings do not twist. It is, of course, better to test each shaft independently, but the error would not be great by using the formula—probably not 1 per cent. either way.

While this instrument was designed primarily for turbine engines, it was quite natural that we should attempt to use it for ordinary piston engines as well. In that case, however, on account of the variable torque throughout a single revolution, not less than six magnets ought to be fitted round the gun-metal wheel, and six sets of inductor sectors. This has actually been done in the case of two small twin-screw vessels with success, but it was a very cumbersome arrangement.

The diagrams in Figs. 3 and 4, give the results of the experiments, from which it will be seen that they are very consistent, and show about 94 per cent. efficiency for the steam engine.

Fig. 6 shows a set of recording instruments for three shafts, and the wheels and sectors for one shaft fitted to a dummy; the actual instruments are exhibited on the table. Fig. 9 shows one of the wheels as actually fitted in a tunnel.

As a still further convenience, we use on all our trials of turbine vessels another machine, shown in Fig. 7, with a roll of paper, on which a time pen works, and also pens recording the revolutions of the turbines; and for convenience, these are only recorded for every tenth revolution of the turbine. This machine is also connected by electric cables with the bridge, by means of which the beginning and the end of the mile can be signalled, and thus we are able, at one recording station, to ascertain the mean speed, the mean revolutions and the mean power, all taken by one or two observers in a quiet and undisturbed cabin.

The diagram in Fig. 5 shows the results got from two practically sister vessels, one fitted with twin screws, and the other with turbines.

The great advantage of these instruments is that we separate and define the efficiency of the turbine and the screw,

and by this means we have been able to discover which is the best combination for any particular ship.

I feel we owe a great deal to the skill and perseverance of Mr. Charles Johnson, a son of the late Mr. Charles Johnson, a member of this society, who was with us for many years, and latterly in partnership with the late Mr. William John.

TORQUE OF PROPELLER SHAFTING.

TORQUE OF PROPELLER SHAFTING: SOME INVESTIGATIONS AND RESULTS.*

By J. HAMILTON GIBSON.

The necessity of ascertaining the horsepower of the marine steam turbine by some means analogous to the "indicator" has revived interest in the torsion meter as applied to propeller shafting.

Indicated horsepower, as obtained by mean-pressure diagrams from a reciprocating engine, is all very well from the naval architect's point of view so long as frictional and other losses are known and remain fairly constant; but the actual power transmitted to the propeller by the shaft is more definitely valuable information, if such is obtainable and can be relied upon.

In utilizing a steel bar, such as a propeller shaft, for the measurement of power, obviously the first thing to do is to "calibrate" that portion of the shafting which is to be so utilized; for the shaft behaves exactly like a steel spring. However accurate the dimensions and homogeneous the material, there are slight differences from the calculated results that must be known; just as there are discrepancies in a set of apparently identical helical springs for boiler safety valves. Each must be tested and set independently; and the analogy suggests that shafts used for torsion-meter records should be recalibrated periodically, just as safety-valve springs are.

A screw-propeller shaft, when transmitting power, undergoes a compound strain. Not only is it twisted by the engine; it is simultaneously compressed by the thrust of the propeller. That this thrust is not an entirely negligible quantity may be inferred from the fact that in an actual case, the torsional

* Paper read before the Institution of Naval Architects, March 21, 1907.

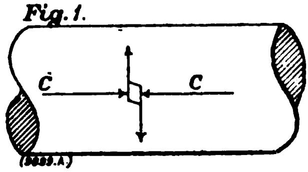
stress in the shafting at full power being 5,100 pounds, the compressive stress due to thrust works out at 950 pounds per square inch, or 18 per cent. of that due to torque alone.

In calibrating the shafting, then, it is advisable to make an allowance for, or to imitate as far as practicable, the actual working conditions, and to fix the power constant accordingly.

Appendix No. I shows by calculation the probable effect of the compound stress referred to; but it is given with all reserve, as actual experiments do not show quite so marked an effect. This is probably due to the fact that, even when transmitting full power, a propeller shaft is rarely twisted more than 1 degree in 10 feet, and the effect of compression will naturally be more and more marked as the twist increases.

Thus a minute square on the surface of a shaft will become skewed and form a rhombus as the shaft twists, and the square will be pushed out of shape more easily by axial compression C, C, Fig. 1, as the shaft goes on twisting. It follows, also,

Fig. 1.



that the effect of compression will be more marked in a hollow shaft than in a solid shaft, and in a shaft having a very large hole than in one of smaller bore; for the skewing effect becomes greater as the surface of the shaft is approached, whilst the core remains comparatively unaffected by the torque, and offers an unimpaired resistance to compression.

Figs. 2, 3, 4 and 5 are descriptive of an actual calibration experiment carried out in accordance with these ideas.

The torque and compression were calculated beforehand for the varying speeds, allowing 95 per cent. of the estimated equivalent indicated horsepower for torque, and assuming 65 per cent. of the same to be re-transmitted from the propeller to the vessel in the form of thrust. In other words, the turbine efficiency was taken as 95 per cent., and the propulsive

efficiency as 65 per cent., giving a propeller loss of 30 per cent. The figures are necessarily somewhat arbitrary, but sufficiently near for the purpose.

Two light pointers were secured to the shaft at a certain fixed distance apart, and of such a length that their points

Fig. 2

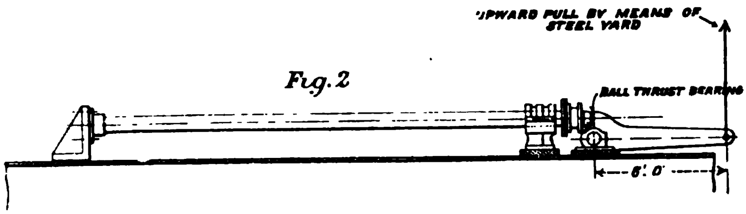


Fig. 3.

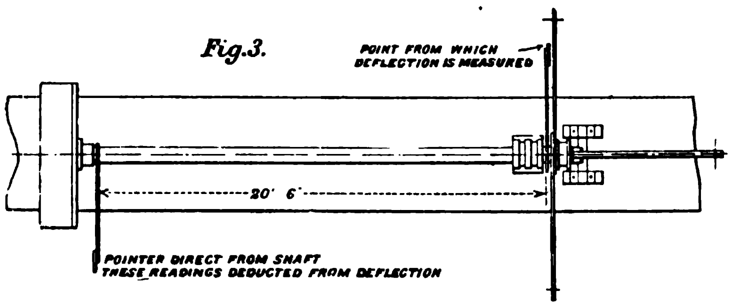
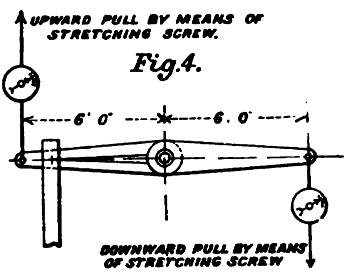


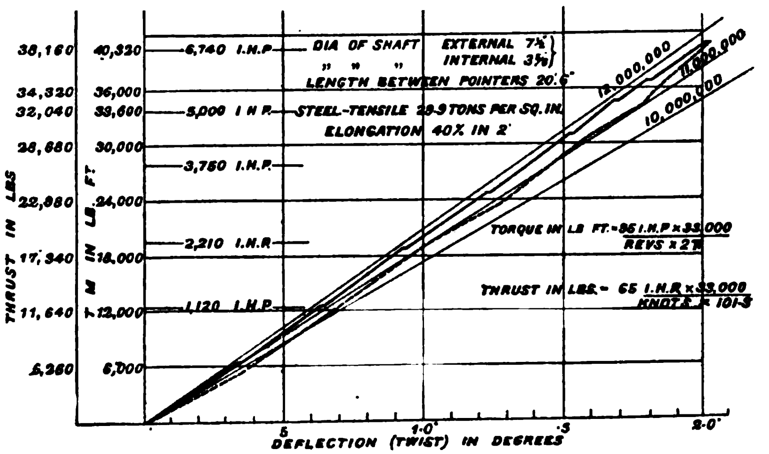
Fig. 4.



described 1 inch of arc for 1 degree of torque. The torque could thus be readily measured by a fitter's decimally-divided straight edge. The difference between the pointer readings gave the exact torque. It is worth notice that, however secure the fast end of the shaft may be made, there is always

Fig. 5 CALIBRATION TEST OF STEEL SHAFTING.

COMBINED TWISTING MOMENT AND CORRESPONDING THRUST T.M. APPLIED FIRST, THEN BEING INCREASED CONTINUOUSLY. RETURN LINE (DOTTED) GIVES THE EFFECT OF DECREASING THE LOADS (T.M AND THRUST) SIMULTANEOUSLY, AND AT THE SAME STAGES AS ON THE UPWARD LINE



Note — TOTAL DEFLECTION DUE TO THRUST ABOUT ·08°= 4% OF THE GROSS DEFLECTION

the liability of some movement, especially at the beginning of an experiment, due to fastenings, such as bolts, keys, &c., giving slightly as the stress comes on.

The torque was applied by spring balances at opposite ends of a couple, to eliminate friction in the supporting bearing, whilst the shaft was compressed by means of a 6 to 1 bell-crank lever loaded by weights on a steelyard. To enable the end of the shaft to turn freely, a ball bearing was introduced between it and the compression lever.

Referring to the diagram in Fig. 5, the loads were applied in the following order:

1. First increment of torque.
2. Corresponding increment of compression.
3. Adjust torque balances..
4. Second increment of torque.
5. Second increment of compression.
6. Adjust torque balances. And so on.

An appreciable time was allowed between each increment for the shaft to accommodate itself to the stress. The loads were taken off in inverse order.

The effect of adding the compression each time was not to increase the angle of torque sensibly and directly, as might be expected, but to slightly relieve the loads on the torque balances. When, however, the torque balances were screwed up again and adjusted to their former load, the shaft was found to be twisted an additional amount, represented by the short horizontal lines on the upward curve, and these horizontal lines became appreciably longer as the full load was attained. The phenomenon of mechanical hysteresis or "lag" was very marked on the return curve, shown dotted; but on the removal of the last increment of load the pointer immediately returned to zero. As a check, the experiment was repeated several times with variations, such as applying the loads quickly, and taking off the total load suddenly; but the curves followed the same lines within narrow limits, and the shaft invariably and instantly returned to zero, thus proving its perfect elasticity, but leaving some doubt in one's mind as to the real nature of the "lag" referred to. Apparently a higher modulus should be taken for a shaft in which the power is gradually and uniformly augmented, and a lower modulus as the power is gradually and uniformly reduced. But these conditions do not obtain in actual work. A constant fluctuation of load giving rise to torsional oscillations in the shaft occurs, which probably has the effect of harmonizing the maximum and minimum torque for any given power; and the true modulus might be taken as a mean between the upward and downward curves. In the present case the average modulus is 11,250,000, giving a constant of 3.27 in the horsepower formula.

Thus

$$H = \frac{\theta D^4 N}{C L} \text{ for solid shafting,}$$

and

$$H = \frac{\theta (D^4 - d^4) N}{C L} \text{ for hollow shafting,}$$

where

$H=$ shaft horsepower.

$\theta=$ angle of torsion in degree.

$D=$ diameter of shaft in inches.

$d=$ diameter of hole in shaft in inches.

$N=$ number of revolutions per minute.

$C=3.27$ (corresponding to a modulus of rigidity of 11,250,000)

$L=$ length of shaft in inches.

In a more recent experiment, the effect of keeping the shaft "alive" during the calibration test was tried with good results. Two lengths of 6½-inch solid-steel shafting were coupled together, as shown in Fig. 6, and were kept "on the dither" by tapping the surface with lead hammers.

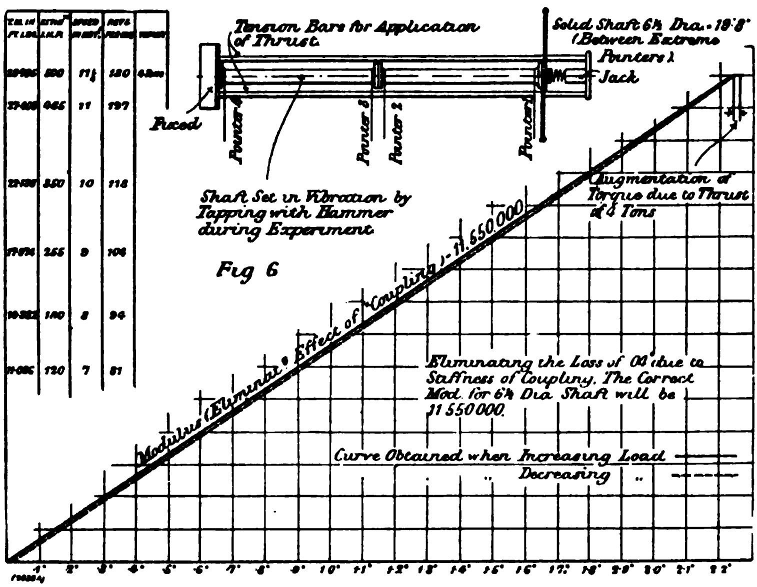
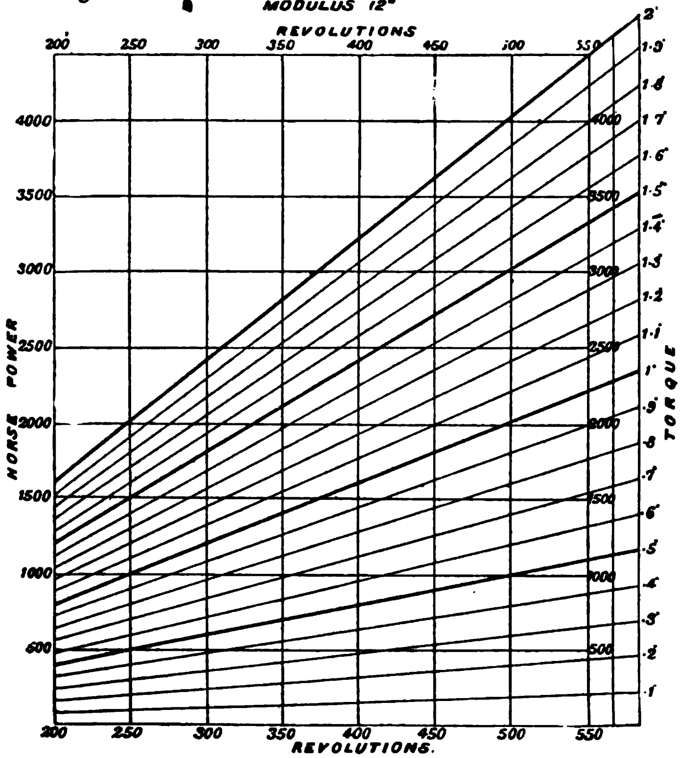


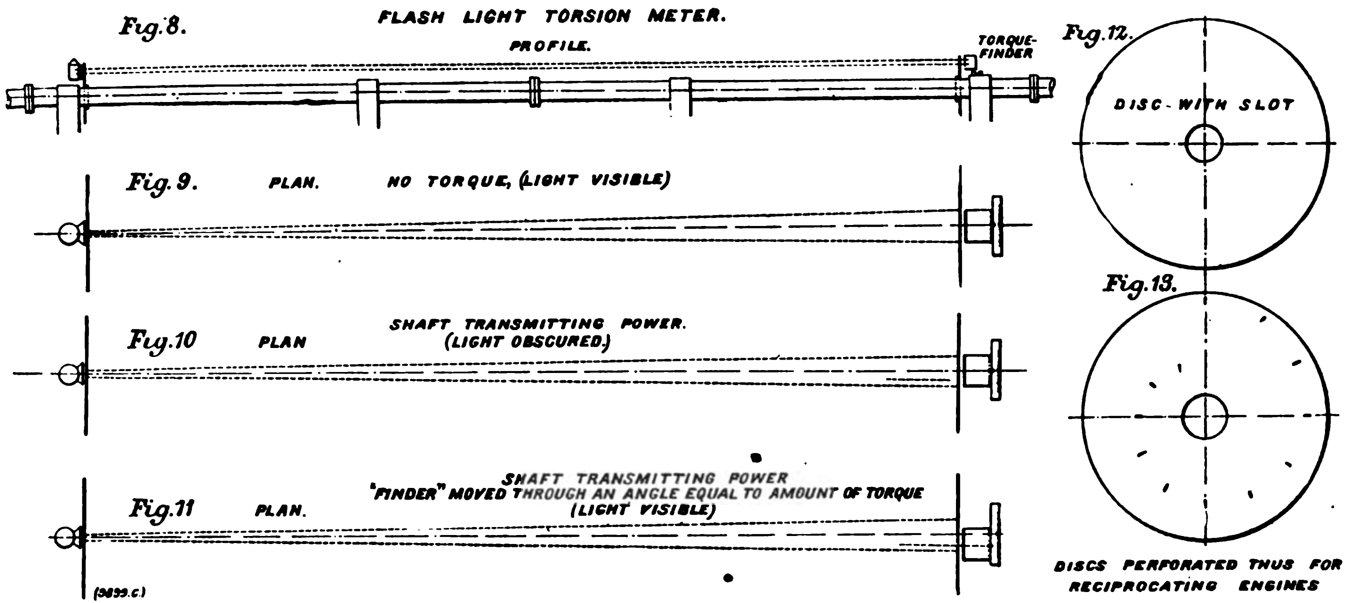
Fig 6

It will be noticed that the lag in this case is almost imperceptible, and that the curve follows a straight line, as, indeed, it should do within the limits of elasticity. Evidently it is important to set up and maintain a species of molecular vibration in making such tests as we are considering. Two

additional pointers were used in this experiment to measure and eliminate the effect of the center coupling, and the compression corresponding to the propeller thrust at full power was added by a screw jack through a calibrated spring and ball bearing after the final increment of torque. The augmentation of torque recorded was just 1 per cent., the torque balances going back ¼ cwt. from 25 cwt. The horizontal line at the top end of the curve shows the additional torque on screwing up the balances again to 25 cwt. The experiments, so far, appear to show that an augmentation of torque due to propeller thrust of from 1 to 1½ per cent. for solid shafting, and from 3 to 4 per cent. for hollow shafting, is a sufficient allowance to make under ordinary working conditions for shafts of normal proportions.

Fig. 7 WING SHAFT 7½ DIA LENGTH 18' 7½"
MODULUS 12"




FLASH LIGHT TORSION METER.
Fig. 8.
PROFILE.
TORQUE-FINDER
Fig. 9.
PLAN.
NO TORQUE, (LIGHT VISIBLE)
Fig. 10
PLAN
SHAFT TRANSMITTING POWER.
(LIGHT OBSCURED.)
Fig. 11
PLAN.
SHAFT TRANSMITTING POWER
"FINDER" MOVED THROUGH AN ANGLE EQUAL TO AMOUNT OF TORQUE
(LIGHT VISIBLE)
(9899.C.)
Fig. 12.
DISC-WITH SLOT
Fig. 13.
DISCS PERFORATED THUS FOR
RECIPROCATING ENGINES

The horsepower formula is such that for a shaft of given diameter, length and ascertained elasticity a diagram as shown in Fig. 7, can be readily constructed to enable the power to be read off at a glance when the revolutions and angle of torque are known. The revolutions are usually readily obtainable by counter or tachometer, and it only remains to ascertain the torque, which must be done with extreme accuracy if the resulting horsepower is to be anywhere near the truth.

Torsion-Meters.—There are two well-known methods in use for measuring the torque of a revolving shaft: one electrical, the other mechanical. An example of the former method is the Denny-Johnson torsion meter. The mechanical method is represented by the Föttinger apparatus. It is not proposed to refer to these methods any further than to say that both involve certain intermediate links between the shaft under observation and the recording instrument, and that such links are liable, under varying conditions, to introduce errors.

The apparatus now to be considered abolishes these intermediate links, and measures the torsion directly from the shaft itself. As a beam of light is the essential feature of the apparatus, it has been called the "Flashlight" torsion meter, and will be so referred to in the following description. The intrinsic parts of the flashlight torsion meter consist of two thin discs secured to and revolving with the shaft, a fixed lamp, and a movable "torque finder." Figs. 8 to 13 show how these are disposed with relation to the shaft, and Figs. 14 and 15 are photographs of the apparatus as actually fitted.

The discs are similar, and each is perforated near its periphery by a small radial slot. The lamp has a mask perforated with a similar slot, and the torque finder has an eyepiece which can be moved circumferentially, and is also provided with a slot similar to the others. When the four slots are in the same radial plane, an observer looking through the eye-piece can see the beam from the lamp, and at every revolution of the shaft a flash is seen. The faster the shaft re-

volves the steadier the light appears, owing to the physical effect of persistence on the retina. The apparatus forms, in fact, a pair of photographic shutters, one shutter being at the lamp end and the other at the torque-finder end.

Immediately the shaft begins to transmit power the discs twist relatively to one another, and their slots are not now contained in the same radial plane. The beam still flashes through the shutter at the lamp end, but is obstructed at the torque finder, because of the relative displacement of the discs. To pick up the flash again it is necessary to move the eye-piece circumferentially by an amount equal to this displacement. Attached to the eye piece is a vernier, which moves over a fixed graduated scale on the torque finder, and enables the angle of torsion to be read off in degrees.

The width of the slots is eliminated by utilizing their edges, Fig. 8, and noting carefully the exact instant that the last ray is cut off as the eye piece is moved over. This gives an infinitesimally fine radial line of light as datum, and enables the torque to be measured to $\frac{1}{100}$ deg., or $\frac{1}{36000}$ of the circumference. No difficulty is experienced in attaining this degree of accuracy. Several independent observers have on occasion obtained exactly the same reading on setting the torque finder for themselves, an operation which takes only a few seconds to perform. To illustrate the appearance of the flash, as viewed through the torque finder, and also the method of measuring the torque and ascertaining the power, an image will now be projected on the lantern screen.

We will suppose, first, that the shaft is revolving idly; the propeller is disconnected; or, if in a turbine vessel, the steam supply may be shut off for a few moments and the turbine opened to the condenser vacuum. The "way" of the vessel will then cause the propeller to revolve the shafting and turbine without appreciable torque. By moving the torque finder over until the flash is just cut off, we obtain the "zero" reading, which must, of course, be subtracted from any subsequent power readings. The zero in this case is, say, 0.56 degree. Now we will suppose that the shaft is transmitting

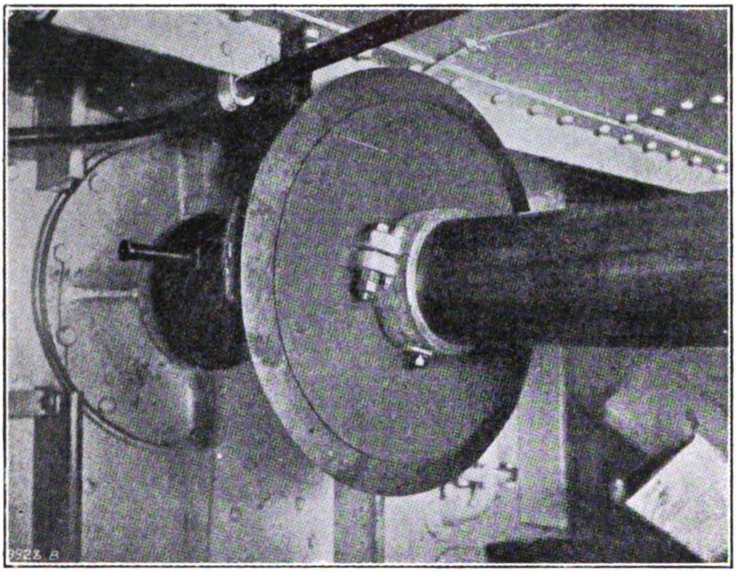


Fig. 14.

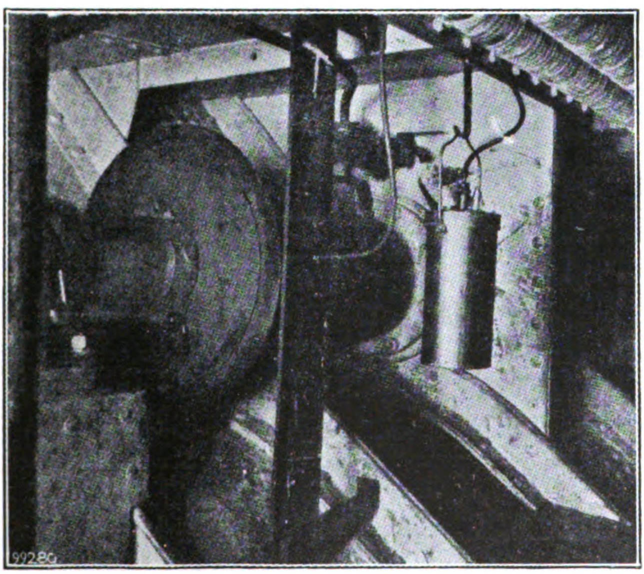


Fig. 15.

Fig. 16.

power, and has twisted the discs relatively to one another, thus obscuring the light. To pick up the flash again, the torque-finder eye piece, with its vernier, must follow the slot in its corresponding disc, and to get the exact angle of torque the point of cut-off is located as before. The vernier scale now shows 2.45 degrees, which, when the zero reading is subtracted, gives an effective torque of 1.89 degrees. To obtain the horsepower we refer to a power diagram similar to that shown in Fig. 7. Assuming that we know the revolutions to be 500 per minute, then in this case the horsepower is seen to be 3,820.

In Appendix II are set out in progressive form some shaft horsepowers recently obtained on a three-shaft turbine steamer by means of the flashlight torsion meter. The results extend over a series of trials carried out under varying conditions, the total powers recorded ranging from 7,975 down to 37. It will be seen that the port low-pressure turbine indicates less power than the starboard throughout the series. On investigation it was found that the blade-tip clearances differed slightly, the percentage difference corresponding almost exactly with the percentage difference of the powers.

In a turbine installation the turning moment is so uniform that it is perhaps only necessary to indicate the torque at one point of the revolution. But in shafting driven by reciprocating engines the turning moment is anything but uniform, and a torque reading taken at only one point in the revolution might give an entirely misleading result. The torque should be ascertained at several points in the circle, and plotted out on a development of the circumference to obtain the mean turning moment.

To meet this condition the flashlight torsion meter is slightly modified, as shown in Fig. 13. Instead of having only one slot at a fixed radius, each disc is perforated with several slots, corresponding to the number of points in the revolution at which it is desired to ascertain the torque, the slots being disposed in the form of a spiral at varying radii. The outermost slot, for instance, may indicate the torque when

the high-pressure crank is on its top center, and so on. The light and the eye piece of the torque finder must be moved radially to or from the shaft to come opposite each pair of slots in the discs, and observations made, as before described, for each position.

A torsion-meter record taken from a reciprocating-engine shaft by this method, and a set of indicator diagrams obtained simultaneously from the engine, would provide an interesting means of comparison, and very shortly such a comparison may be available. Of perhaps greater interest would be a comparison between the torsion-meter records of two identical vessels run at the same speed, one of which is propelled by turbines, the other by reciprocating engines; as, for example, in the Cunard liners, *Carmania* and *Caronia*, or H.M. S. *Amethyst* and *Topaze*. We should then have the exact power transmitted to the propellers of each vessel respectively, the only disturbing element in an exact comparison being the vexed question of propeller efficiency. But a consideration of the compound stresses in propeller shafting suggests that this problem might be solved by the use of some simple apparatus that would record the actual thrust along the line of shafting, and enable a direct comparison to be instituted between the power transmitted to, and the thrust produced by, the propeller. This, however, would naturally form the subject of another paper.

NOTE.—Besides the figures and diagrams appended, several lantern slides, some being mechanical, were projected on the screen, and added considerably in conveying to the minds of the audience the principles and methods of working the flash-light torsion meter. A working model of the apparatus was on exhibition, together with actual specimens of discs, lamps, torque finders, &c., all of which evoked the interest of members present.

But the above applies only to the outer skin of the shaft when the twisting stress is a maximum (Fig. 18). The twisting stress at the center is *nil*, whilst the compression stress is uniformly distributed over the whole area of section. There-

fore the effect of compression in weakening the shaft's resistance to torsion will be less than the percentage above quoted.

Fig. 17.

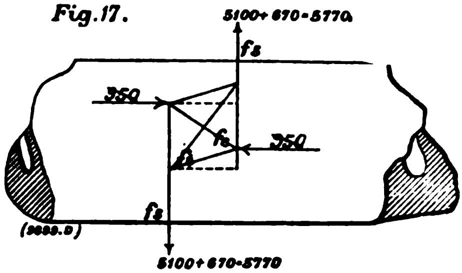
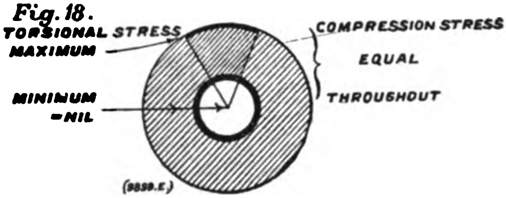


Fig. 18.



If the effect at the surface is 11½ per cent., and at the core *nil*, we may assume a mean effect over the whole section to be 0.5 of 11½ per cent. = 5¾ per cent. for a solid shaft; and 0.75 of 11½ per cent. for the hollow shaft in question = 8.6 per cent. for a hollow shaft of normal proportions. The true modulus of rigidity for calculating power would then be 8.6 per cent. less than modulus for torsion alone. Actual experiments, however, show only about half of this effect (see diagram, Fig. 5).

APPENDIX I.

Investigation for Torsion Test of Shafting as Described in Figs. 2 to 5.

Shaft, $7\frac{1}{2}$-in. in external diameter.
$3\frac{5}{8}$-in. hole.
Length between pointers, 20 feet 6 inches.

Maximum deflection in degrees

$$=\frac{H.P. \times C \times L''}{(D^4-d^4) \times \text{revs.}}=\frac{5{,}000 \times 0.95 \times 3.27 \times 246}{2{,}997 \times 750}=1.7 \text{ degree.}$$

($C = 3.27$ for a modulus of 11,250,000),

and maximum thrust

$$=\frac{0.65 \times 5{,}000\ H.P. \times 33{,}000}{33 \text{ kts.} \times 101.3 \times 2{,}240}=14.3 \text{ tons.}$$

Area of shaft $= 33.7$ sq. inches: $\frac{14.3 \times 2{,}240}{33.7} = 950$ pounds per square inch compressive stress.

Torsional (or shear) stress at full power—

$$T.\ M. =\frac{33{,}000 \times 0.95 \times 5{,}000\ H.P.}{750 \text{ revs.} \times 2\pi} = 3{,}300 \text{ lbs. at } 1' \text{ radius.}$$

$$T.\ M. = 0.196\left(\frac{D^4-d^4}{D}\right)f, \text{ therefore } f = \frac{33{,}300 \times 12 \times 7.5}{0.196 \times 2{,}997}$$

$= 5{,}100$ pounds per square inch torsional stress.

A shear stress produces equal tension and compression stresses at 45 degrees–*i. e.*, $f_s = f_t = f_c$. (Lineham.)

But the compressive stress due to propeller thrust augments the

$$f_c \text{ by } \frac{950}{\sqrt{2}} = 670 \text{ pounds per square inch,}$$

and as $f_s = f_c$, therefore 670 must be added to the $5{,}100+670 = 5{,}770 =$ total $f_s =$ gross torsional (or shear) stress, and $670 = 11\frac{1}{2}$ per cent. of 5,770 (Fig. 17).

APPENDIX II.

Flashlight Torsion Meter.—Turbine Steamer.

Actual readings and corresponding horsepowers taken during trial trips under varying conditions of displacement and propellers.

Turbine shaft.	Degrees torque.	Revolutions per minute.	Shaft horsepower.	Shaft horsepower. Total.
Starboard low-pressure	1.43	482.9	2,775	
Center high-pressure	1.69	461.2	2,600	7,975
Port low-pressure	1.37	472.8	2,600	
Starboard low-pressure	1.31	461.2	2,410	
Center high-pressure	1.65	426.8	2,330	6,940
Port low-pressure	1.24	457.3	2,200	
Starboard low-pressure	1.15	426.4	1,970	
Center high-pressure	1.51	417.6	2,080	5,960
Port low-pressure	1.13	418.9	1,910	
Starboard low-pressure	1.05	418.4	1,765	
Center high-pressure	1.52	422.3	2,120	5,555
Port low-pressure	1.02	415.5	1,670	
Starboard low-pressure	0.21	198.6	162	
Center high-pressure	0.27	206.3	185	495
Port low-pressure	0.19	183.5	148	
Starboard low-pressure	0.22	146.7	88	
Center high-pressure	0.21	171.4	87	257
Port low-pressure	0.13	144.8	82	
Starboard low-pressure	0.07	46.3	13	
Center high-pressure	0.05	86.1	15	37.2
Port low-pressure	0.01	24.4	9.2	

U. S. S. *KANSAS.*

DESCRIPTION AND OFFICIAL TRIALS.

By WILLIAM ASHLEY LEAVITT, JR., ASSOCIATE.

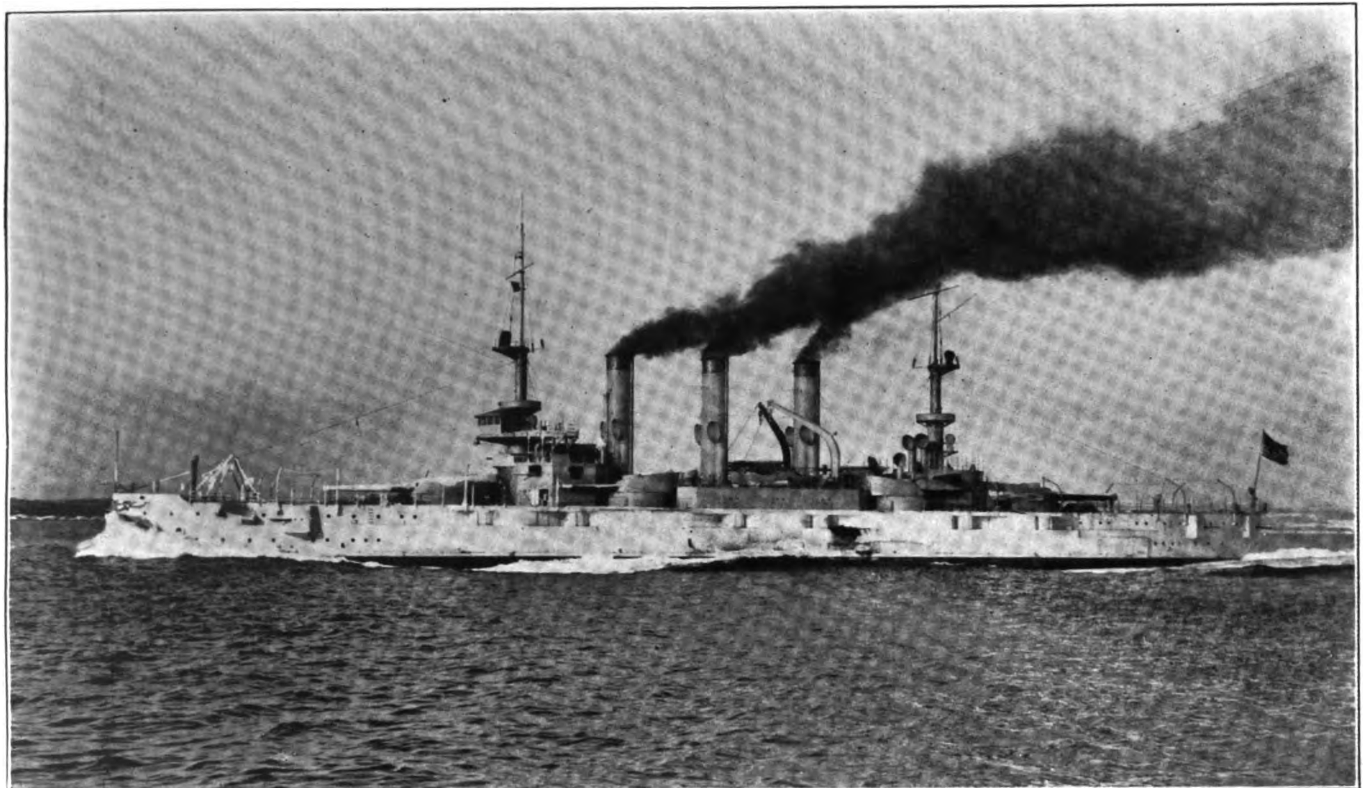
The *Kansas* (Battleship No. 21) building by The New York Shipbuilding Company, Camden, New Jersey, is one of three ships of this class authorized by Act of Congress, March 3d, 1903, the sister ships being the *Minnesota* and *Vermont.*

The contract was signed July 16th, 1903; keel laid February 10th, 1904; hull launched August 12, 1905; and the vessel was delivered to the Government April 15, or four months over the contract time, forty-two (42) months; the contractors claiming non-receipt of armor and certain details of ordnance as the principal causes of delay.

The contract price for the construction of hull and machinery was $4,165,000, of which sum $3,165,000 was allotted to hull and $1,000,000 for the propelling machinery and auxiliaries. The contract required a speed of 18 knots to be maintained for four consecutive hours on a mean draught of 24 feet 6 inches, corresponding to a displacement of 16,000 tons, and an additional endurance run for twenty-four consecutive hours at not less than 13,200 I.H.P., equivalent to eighty per cent. of the designed H.P.

The contract allotment of weights for machinery was 1,500 tons; the actual weight being about 1,504 tons, or about 4 tons overweight.

The propelling machinery was given dock trials as follows: Starboard engine, November 12th and 13th, 1906; aggregate number of hours run, 16; maximum revolutions, 77. Port engine, November 14th, 15th, 16th, 1906; aggregate number of hours run, 24; maximum revolutions, 78; the results being



U. S. S. "Kansas."

generally satisfactory, the only alterations of note being the addition of heavy cast-steel brackets resting on the inner bottom to support the overhang of the long main bearings between the I.P. and after L.P. cranks, and an increase in the size of the relief valves on the main feed discharge.

OFFICIAL TRIALS.

The Navy Department having given permission to begin the official trials, December 13th, 1906, the ship left Camden at 1 P. M., December 5th, proceeding to Boston for docking at the Navy Yard, reaching that place at 8 A. M., December 9th, or 91 hours out from Camden. Fog in the Bays at both ends of the trip, and a strong head wind with a fairly heavy sea running being the principal causes of delay. The ship was in dock exactly fifty hours, leaving for Rockland at 6 A. M., December 12th, and anchoring off Owls Head Light at the entrance to Rockland Harbor at 4·30 P. M. This run being made under the first favorable weather conditions experienced since leaving Camden at 2 P. M., a run at maximum power was begun, but after running 55 minutes it was called off on account of the after main bearing of the port engine showing signs of heating. This was the only speed run attempted by the contractors before the Trial Board took charge and the official trials were made. December 13th being favorable both as regards wind and sea, the standardization runs were made over the Government measured-mile course off Monhegan Island, the maximum speed attained being at the rate of 18.499 knots at 122.55 revolutions, developing 20,373 I.H.P.; and the average of the five runs at full power was 18.094 knots at 121.43 revolutions, developing 19,410 I.H.P.

On December 14th the four-hour endurance run in open sea was made successfully, the course being laid along the Maine coast, and headed toward Boston Light. The most remarkably close engine running was maintained throughout the four hours, the average variation in the revolutions being less than one-half of a revolution. The average number of

revolutions for the starboard engine was 122.12; port engine, 130.52, and average of both engines, 121.32, developing 1,930 I.H.P. for the main engines, or a total for all machinery of 19,757 I.H.P. The average speed was 18.094 knots, being exactly the same as the average of the five full-power standardization runs; average slip of the propellers being 14.86.

The performance of the machinery, both main and auxiliary, was generally satisfactory, except in details as follows: The port after main bearing had to be watched very closely, as it gave frequent evidences of heating, and an excessive amount of oil had to be used on all bearings, due, undoubtedly, to the excessive load and strains on the working parts on account of the engines being called upon to develop about 20 per cent. above the designed power.

The port H.P. and forward L.P. crank pins heated during the last 15 minutes, requiring the use of water, and the H.P. and I.P. crank pins and crossheads of both engines pounded considerably. When called upon to deliver considerable feed the main feed pumps pounded a good deal, which was undoubtedly due to the suction pipes being too small.

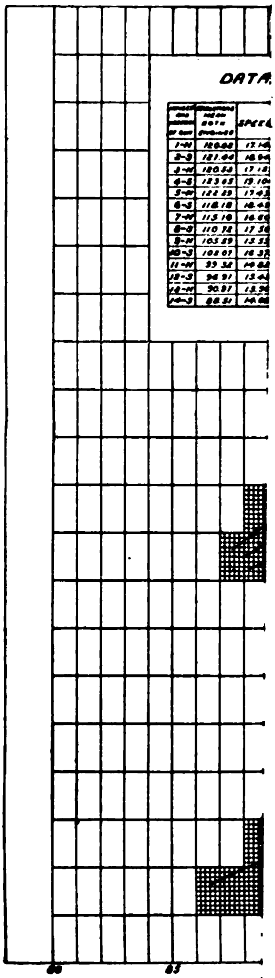
STANDARDIZATION.

For the purpose of standardizing the screws, runs were made over the measured-mile course off Rockland, Maine, Dec. 13, 1906. The weather was clear and cool with moderate to gentle breezes. Smooth sea.

Previous to the vessel's starting on the standardization runs the draught was taken and found to be—

Forward, feet and inches, ...	24–08½
Aft, feet and inches, ...	24–05⅝
Mean, feet and inches, ...	24–07$\frac{1}{16}$
Corresponding displacement, tons, . . .	16,085

These runs gave the data from which the curves on Plate I were plotted.



	Revolutions Mean Both Engines	SPEE
1-N	120.08	[illegible]
2-S	121.04	[illegible]
3-N	120.53	[illegible]
4-S	123.05	[illegible]
5-N	122.29	[illegible]
6-S	118.18	[illegible]
7-N	115.10	[illegible]
8-S	110.72	[illegible]
9-N	105.59	[illegible]
10-S	102.01	[illegible]
11-N	99.32	[illegible]
12-S	96.91	[illegible]
13-N	90.37	[illegible]
14-S	88.31	[illegible]

After the standardization run the draught was—

Forward, feet and inches, ...	24–07$\frac{3}{8}$
Aft, feet and inches, ...	24–05$\frac{7}{8}$
Mean, feet and inches, ...	24–06$\frac{5}{8}$
Corresponding displacement, tons, . . .	16,064

FOUR-HOURS' OFFICIAL TRIAL.

The ship left her anchorage at Rockland, Maine, at 8·00 A. M. December 14, 1906, and steered a straight southeast course out to sea. A period of two hours was consumed in working the machinery up to power, and at 9·51 the signal was given that the four-hour run was begun, which was finished at 1·51 P. M. without the least trouble arising to cause a pause for an instant. The main engines ran with remarkably little vibration, the smooth running of the valve gear, which seemed to almost float, being particularly noticeable. The water service being used only on the crosshead guides and thrust bearings, furnishes satisfactory evidence of good running and workmanship.

At the conclusion of the four-hour run the engines were slowed down for maneuvering the ship. The time required to put the helm from hard-a-starboard to hard-a-port was ten seconds, and from hard-a-port to hard-a-starboard was ten seconds. During this test the maximum angle of heel was seven degrees.

In the firerooms during the four-hour run everything worked easily and with great regularity. The time-firing device installed by the contractors, the design of two men in their employ, regulated the intervals of firing, which averaged five minutes for each furnace. The performance of this device was most satisfactory and the simplicity of design makes it easy to regulate, overhaul and stow in the ship. The forced-draft blowers and engines ran with noted smoothness, and at no time did the revolutions exceed 340 per minute. In the foreward firerooms at intervals the auxiliary feed pumps were used to make up feed, the main pumps not being able to

reach those boilers when forced. In order to determine accurately the amount of coal consumed it was put into bags holding 200 pounds each, and a record kept of the number of bags used.

The number of men used in each fireroom was as follows:

1 Water tender.
4 Firemen.
2 Coal passers in fireroom.
2 Coal passers in bunkers.
1 Apprentice boy.

DATA FOR FOUR-HOUR RUN.

Draught at beginning of trial, forward, feet and inches	24-07$\frac{1}{8}$	
aft, feet and inches	24-07$\frac{3}{8}$	
Mean draught at beginning of trial, feet and inches	24-07$\frac{1}{4}$	
Corresponding displacement at mean draught at beginning of trial, tons	16,103	
Mean draught at end of trial, not taken, feet and inches, (5·00 P. M.)	24-05$\frac{1}{2}$	
Corresponding displacement at mean draught during trial, tons	16,000	
Speed of ship, in knots per hour		18.094
	Starboard.	*Port.*
Slip of propellers, in per cent. of their speed	15.43	14.29

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures (*Average of one-half hourly observations.*)

	Starboard.		*Port.*
Steam pressure at boilers (per gauge), pounds		278.2	
engine (per gauge), pounds		264.5	
H.P. steam chest (gauge), pounds	250.0		250.0
1st receiver (absolute), pounds	105.0		108.0
2d receiver (absolute), pounds	38.0		38.0
Vacuum in condensers, inches of mercury	28.0		27.9

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	44.0		44.4
Discharge, degrees	101.0		101.6
Hotwell, degrees	69.0		69.2
Feed water, degrees	165.0		164.4
Engine room, upper platform, degrees	86.0		89.5
working platform, degrees	74.0		74.0
Firerooms, above grating, degrees		68.0	
working level, degrees		65.7	
Atmosphere		32.5	

Revolutions, or double strokes, per minute. (Average of one-half hourly observations.)

	Starboard.	Port
Main engines (mean for trial)	122.12	120.52
average for both	121.32	
Pumps, main, air	19.0	18.0
circulating	140.0	133.0
feed	27.0	32.0
bilge	64.0	43.0
auxiliary feed	......	......
condenser	13.0	36.0
Blower engines	314.0	
Speed of ship, in knots per hour	18.004	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	15.43	14.20
Air pressure in firerooms, inches of water	.596	

Mean Effective Pressures in Cylinders, in pounds per square inch. (Averages of cards taken at half-hourly periods.)

	Starboard.	Port
Main engines, H.P. cylinder	123.5	122.5
I.P. cylinder	49.4	49.8
F.L.P. cylinder	20.7	20.2
A.L.P. cylinder	20.9	21.0
Mean equivalent pressure, in pounds per square inch referred to combined area of L.P. pistons	56.61	56.46

INDICATED HORSEPOWER.

	Starboard.	Port
Main engines, H.P. cylinder	2,958.0	2,896.0
I.P. cylinder	3,194.0	3,183.0
F.L.P. cylinder	1,778.0	1,711.0
A.L.P. cylinder	1,797.0	1,785.0
Total	9,727.0	9,575.0
both engines	19,302.0	

AUXILIARIES.

	Starboard.	Port
Air pumps	13.7	13.3
Circulating pumps, main	52.3	44.5
Feed pumps, main	55.6	64.1
Bilge pumps	9.8	3.9
Water-service pumps	...	...
Auxiliary condenser pumps	1.4	4.1
Blower engines (forced draft)	97.6	
Steering engines (estimated)	5.0	
Dynamo engines	90.0	
Total	455.0	
Total all machinery in operation	19,757.0	

COAL.

Kind and quality	Pocahontas, hand-picked, excellent.
Average thickness of fires, inches	10–12
Coal burned per hour, pounds	34,328.0
Average temperature of atmosphere, degrees Fahrenheit	32.5
air pressure in firerooms, inch of water	.596

Deduced Data.

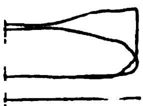
Coal burned per square foot of G.S. per hour, pounds	31.21
H.S. per hour, pound	.651
I.H.P. of main engines per hour, pounds	1.779
total I.H.P. main engines and all auxiliary engines per hour, pounds	1.737

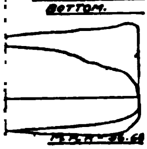
INDICATOR CARDS.

One set of cards representing maximum conditions obtainable under forced draft without steam to receivers were selected for comparison with theoretical cards and average condition on the trial. See the accompanying sketch of the cards and the same combined, Plate II.

Port Engine.	Cards taken witho't steam to receivers. M.E.P.	Cards taken on trial with steam to receivers. M.E.P.
H.P. cylinder, average	125.0	....
I.P. cylinder, average	50.15	....
F.L.P. cylinder, average	20.35	....
A.L.P. cylinder, average	21.39	....
Starboard Engine.		
H.P. cylinder, average	125.3	....
I.P. cylinder, average	48.65	....
F.L.P. cylinder, average	20.79	....
A.L.P. cylinder, average	21.01	....

	H.P.	I.P.	F.L.P., A.L.P. Combined.
Area of piston, top, square inches	829.57	2,206.19	5,844.94
Ratio of areas	1.0	2.66	7.05
Per cent. cut off, top, act. average	81.4	78.25	75.75
bottom, act. average	74.25	70.55	64.42
clearance, top, act. average	23.82	17.43	15.32
bottom, act. average	21.81	19.42	17.47

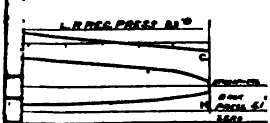




NOTE I
CARDS WERE TAKEN BY THE TRIAL BOARD ON THE 4 HOUR OFFICIAL TRIAL DEC 14. 06. THROTTLE FULL OPEN. LINKS FULL GEAR. BY-PASS CLOSED.

…OR CARDS.
ENGINE.
BOTTOM.
…N 4 HOUR OFFICIAL TRIAL
…N = 1222
…3002
…3142
…3696
…9761
H.P. = 72.5 %
I.P. = 74.3 %
L.P. = 68.9 %
…D ON DIAGRAM.
…RDS = 17.41 □"
…AL CARD A.B.C.H.K. = 30.67 □"
56.6
…S 1" = 30 lbs
… 1 = 30000 CU. INS.
…ED IN EACH CYLINDER
…0 %
…31 %
…89 %
…00 %

SIZE OF ENG. = $\frac{32\frac{1}{2}" \times 53" \times 2 - 61"}{48"}$
RATIO OF CYLINDERS 1 : 2.66 : 7.04.
EXPANSIONS 8.04



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Volume at cut off, top, H.P. cylinder, cubic inches	32,420.0
of clearance, top, H.P. cylinder, cubic inches	9,471.68
at cut off, total H.P. cylinder, cubic inches	41,891.68
bottom H.P. cylinder, cubic inches	28,150.0
of clearance, bottom, H.P. cylinder, cubic inches	8,256.11
at cut off, total, H.P. cylinder, cubic inches	36,406.11
Mean of top and bottom volumes H.P. cylinder, cubic inches	39,148.89
Piston displacement and clearance, top, L.P. cylinders, cubic inches	323,531.78
Piston displacement and clearance, bottom, L.P. cylinders, cubic inches	224,898.17
Piston displacement and clearance, mean, L.P. cylinders, cubic inches	324,214.97
Ratio of initial to final volumes of steam	8.28
Areas of actual trial cards	17.5
theoretical cards	30.55
Ratio of actual to theoretical	.5729

TWENTY-FOUR HOURS' OFFICIAL TRIAL.

On December 15th, at 7 A. M., the ship got under way for the twenty-four-hour endurance run at sea, and the run was begun at 8·20 o'clock, ending at that hour December 16th. The course was from Boston to Cape May, and conditions as regards wind and sea very favorable.

The average speed during the entire run was 16.65 knots, at 109.53 revolutions and 14,532 I.H.P., with a total of 14,532 I.H.P. for all machinery. The draught (estimated) was 24 feet, and the displacement 15,620 tons. Mean slip of propellers, 13.24. Except as noted, the performance of the machinery was very satisfactory. During the last hour the port I.P. crank-pin brasses heated, requiring the use of water, water also being used on the valve guides, I.P. crank pin, as a precautionary measure. Boiler "K" was cut out during the last two hours, on account of a leak in the front cross-connection box. The leak was due to faulty welding, and the box was replaced by the Babcock & Wilcox Co.

DATA OF TWENTY-FOUR HOURS' ENDURANCE TRIAL FROM BOSTON, MASS., TO DELAWARE BREAKWATER, DECEMBER 15, 1906.

	Starboard.	*Port.*
Maximum average revolutions per minute for 15-minute period	112.1	112.6
Average revolutions per minute for 24 hours	109.7	109.36
Mean revolutions per minute for both engines	109.53	

	Starboard.	*Port.*
Maximum steam pressure at boilers, pounds	273.0	
Average steam pressure at boilers, pounds	263.0	
Maximum steam pressure at engines, pounds	253.0	254.0
Average steam pressure at engines (H.P. steam chest), pounds	211.0	220.0
Maximum air pressure in firerooms, inch	.8	
Average air pressure in firerooms, inch	.52	
Maximum vacuum, each engine, inches	27.0	28.0
Average vacuum, each engine, inches	26.8	28.0
Collective I.H.P. of all main engines	7,063	7,091
main engine, air, circulating feed and hot-well pumps	14,332	
Collective I.H.P. of main and auxiliary engines during trial	14,532	
Kind and quality of coal used on trial*	Pocahontas, run of mine.	
Draught at beginning of trial (forward), feet and inches,	24–2½	
(aft), feet and inches	24–4	
Mean draught at beginning of trial, feet and inches	24–3¼	
Corresponding displacement at mean draught at beginning of trial, tons	15,810	
Draught at end of trial	Not taken, ship at sea.	
Speed of ship, knots per hour	16.65	
Slip of propellers, in per centum of their own speed	13.35	13.13

INSPECTION AFTER TRIAL.

SUMMARY OF THE FINDINGS OF THE BOARD.

Starboard Engine.—Examined Nos. 3, 5 and 6. Main bearings and all caps in good condition except No. 5, which was slightly scored and dragged, and No. 6 cap dragged and white metal slightly cracked at the crown. Rolled out bottom brass of No. 6 and found white metal slightly dragged and journal roughened at forward end.

Examined steam and water ends of main air pump; steam end good, but water end showed considerable grease.

Examined steam and water ends of main feed pumps and found them in good condition.

Examined H.P. and I.P. and forward L.P. crank pins and brasses and found them in good condition.

Examined feed tank and found it clean.

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* Average quantity of coal used per I.H.P. per hour not obtainable, there being no observers.

Examined after L.P. crosshead brasses and pins and found them in good condition.

Examined main condensers, which were found to be tight; and fresh-water side, so far as could be seen, was clean, but a few tubes were partly filled with dirt.

Examined all main engine cylinders and H.P. ring and springs and fonnd them in good condition.

Examined H.P., I.P. and forward L.P. valves and found them in good condition.

Port Engine.—The following parts were examined and found in good condition: Main engine cylinders, H.P. valve chest and valve; I.P. valve chest and valves; A.L.P. valve chest and valves, except the H.P. cylinder's, which was cut in one place near the top, and apparently corresponding with a porous spot on the piston ring. The main condenser, feed and filter tank, main air pump, and main feed pumps were found in good condition.

The A.L.P. crosshead brasses and pin were examined and found in good condition. The H.P., I.P. and A.L.P. crank pins were examined and found as follows: The H.P. crank-pin brasses show signs of slight dragging of the white metal, and the white metal is slightly cracked in the crown of the crown brass. The I.P. pin heated on the twenty-four hour run; the white metal in the brasses shows signs of running and dragging, and the A.L.P. pin brasses do not fit properly.

All pins in good condition.

Crank-shaft bearings Nos. 3, 5 and 6, were examined and found in good condition, except No. 6 bearing, which was found slightly cut at the after end for about 1½ inches. This bearing gave considerable trouble previous to the trials, and was refitted twice.

The Board decided that the H.P. pistons should have new packing rings, made solid and without springs; that all crank pins, crank-shaft bearings and crossheads should be readjusted; that the I.P. crank-pin brasses should be relined with white metal; that No. 6 crank-shaft bearing should be refitted, and

that all crank pins and journals should be stoned in places where nicks and scores show.

The steam cylinders and bearings of one blower engine were examined and found in excellent condition; and, as no trouble of any kind was experienced, even at high speed during the trials, examination of the other engines was not considered necessary.

All furnaces and drums or the boilers were carefully inspected; and found in good condition.

Tubes (4-inch) above the furnaces were generally bowed, the maximum bend being about $\frac{5}{16}$-inch degree, bends being upward as well as downward, but a number of tubes were straight.

Evidence of oil, to a very limited extent, was found in all of the boilers.

Five 4-inch tubes which carry the horizontal baffles in the outboard furnace showed signs of bulging, and the Board recommended that they be cut out.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet and inches	450- 0
on load water line (24 feet, 6 inches), feet and inches	450- 0
over all, feet and inches	456-04
of straight keel, feet and inches	362- 0
Projection forward of F. P., feet and inches	6-04
aft of A. P., feet and inches	0.00
Breadth, extreme, at L. W. L., outside of armor, feet and inches	76-10
molded, feet and inches	76-05½
Depth, molded, main deck, at side, M. S., feet and inches	43-01⅝
upper deck, at side, M. S., feet and inches	50-07⅝
Ratio of length to beam	5.94
Draught, L. W. L., for calculations, feet and inches	24-06
mean, on trial, feet and inches	24-05½
Displacement, normal draught, tons of S. W	16,003
mean, on trial, tons of S. W	16,000
per inch, at L. W. L., tons of S. W	63.2
Area of midship section, square feet	1,811
L. W. L. plane, square feet	26,590
wetted surface, square feet	44,000

Center of gravity of L. W. L. plane, aft of M. S., feet	3.39
buoyancy above base line, feet and inches	13-04
aft of M. S., feet and inches	2-08
Transverse metacenter above center of buoyancy, feet and inches	18-09
Longitudinal metacenter above center of buoyancy, feet and in	532-0
Coefficient of fineness, block	.652
M. S	.962
L. W. L	.769
cylindrical	.687
Number of frames	112
watertight compartments	324

Heights above L. W. L. (which is 24 feet 4⅝ inches above the molded base, the molded base line being 1⅝ inches above bottom of keel).

Top of truck light, foremast and mainmast, feet and inches	131- 0
Floor of range-finder platform, foremast and mainmast, feet, and inches	54-11
Searchlight platform, foremast, feet and inches	76- 1
mainmast, lower, feet and inches	43- 2½
upper, feet and inches	76- 1
Conning tower top at center line, feet and inches	42- 2
Flying bridge at side (top of plank), feet and inches	44- 1½
Forward bridge at side, top of beams, feet and inches	34- 1½
After bridge at side, top of beams, feet and inches	33-10½
Top of main-deck planking at stern, feet and inches	20-10½
breakwater at stern, feet and inches	23- 6
beading at stern, feet and inches	22- 1
main-deck planking at stern, feet and inches	20- 4½
flat armor deck, feet and inches	3- 0

MAIN BATTERY.

No. of Guns.	Caliber.	Location.		Train.		Gun No.		Remarks.
		Deck.	F'me.			S.	P.	
				Degrees.	*Degrees.*			
2	12-inch B.L.R.	Main	29	135 S.	135 P.	1	2	In turret.
2	12-inch B.L.R.	Main	85½	135 S.	135 P.	4	3	In turret.
4	8-inch B.L.R.	Gun...	44½	90 F.	55 A.	1-2	3-4	In turret.
4	8-inch B.L.R.	Gun...	69	90 A.	55 F.	5-6	7-8	In turret.
2	7-inch R.F.G.	Gun...	31	90 F.	26 A.	1	2	In sponson.
2	7-inch R.F.G.	Gun...	39	73½ F.	42½ A.	3	4	In sponson.
2	7-inch R.F.G.	Gun...	53	70 F.	46 A.	5	6	In sponson.
2	7-inch R.F.G.	Gun...	61	46 F.	70 A.	7	8	In sponson.
2	7-inch R.F.G.	Gun..	75	42½ F.	73½ A.	9	10	In sponson.
2	7-inch R.F.G.	Gun...	83	26 F.	90 A.	11	12	In sponson.

SECONDARY BATTERY.

No. of Guns.	Caliber.	Location.		Train.		Gun No.		Remarks.
		Deck.	F'me.			S.	P.	
				Degrees.	*Degrees.*			
2	3-inch R.F.G.	Gun...	16¼	90 F.	30 A.	1	2	In sponson.
2	3-inch R.F.G.	Gun...	101	30 F.	90 A.	3	4	In sponson.
2	3-inch R.F.G.	Gun...	109	30 F.	90 A.	5	6	In sponson.
2	3-inch R.F.G.	Main..	52½	60 F.	60 A.	7	8	In casem'e.
2	3-inch R.F.G.	Main..	56½	60 F.	60 A.	9	10	In casem'e.
2	3-inch R.F.G.	Main..	61	60 F.	60 A.	11	12	In casem'e.
2	3-inch R.F.G.	Up'r...	36½	50 F.	50 A.	13	14	...
2	3-inch R.F.G.	Up'r...	50	50 F.	50 A.	15	16	...
2	3-inch R.F.G.	Up'r...	77½	50 F.	50 A.	17	18	...
2	3-inch R.F.G.	F'd bdg.	37	50 F.	50 A.	19	20	...
2	3-pdr. R.F.G....	Up'r ..	39½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Up'r...	75½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	F'd bdg.	40½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	77½	50 F.	50 A.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	79½	50 S.	50 P.	...	...	...
2	3-pdr. R.F.G....	Aft. bdg.	75°	50 F.	50 A.	...	...	...
2	30-caliber.........	F'd bdg.	...	...	...	...	...	...

COAL-BUNKER CAPACITIES.

Measured at 6 inches below bottom of deck beams, and at 43 cu. ft. per ton.

Bunkers.	Location.		Tons.
	Frame.	Side.	
Above protective deck.			
B-119	40½-46	S.	61.9
B-118	40½-46	P.	61.9
B-121	46 -52	S.	70.6
B-120	46 -52	P.	70.6
B-123	52 -58½	S.	77.9
B-122	52 -58½	P.	77.9
B-127	58½-65	S.	76.9
B-126	58½-65	P.	76.9
B-129	65 -72	S.	78.2
B-128	65 -72	P.	78.2
Total....................			731.0
Below protective deck.			
B- 7	35 -40½	S.	120.0
B- 8	35 -40½	P.	131.7
B- 9	35 -40½	S.	50.5
B-10	35 -40½	P.	50.5
B-11	40½-45	S.	50.4
B-12	40½-45	P.	50.4
B-13	45 -49½	S.	69.4

Bunkers.	Location.		Tons.
	Frame.	Side.	
B-14	45 -49½	P.	69.4
B-15	49½-54	S.	75.5
B-16	49½-54	P.	75.5
B-17	54 -58½	S.	77.5
B-18	54 -58½	P.	77.5
B-19	58½-63	S.	77.5
B-20	58½-63	P.	77.5
B-21	63 -67½	S.	76.0
B-22	63 -67½	P.	76.0
C- 5	67½-72	S.	59.2
C- 6	67½-72	P.	73.0
C- 7	67½-72	S.	49.3
C- 8	67½-72	P.	49.3
C- 9	72 -77	S.	51.5
C-10	72 -77	P.	49.5
C-11	77 -83	S.	46.2
C-12	77 -83	P.	46.2
Total....................			1,629.5

Recapitulation, Bunker Capacity.

Above protective deck........	731.0
Below protective deck........	1,629.5
Total....................	2,360.5

TANKS.

Capacities of Fresh-Water Tanks.

Tank No.	Compart. number.	Between frames.	Location.	Side.	Capacity. Gallons.	Tons.
9	D-11	98-101	Hold, aft	S.	2,230	8.3
10	D-11	98-101	Hold, aft	P.	2,230	8.3
1	A-47	20-24	Upper platform, ford.	S.	1,240	4.6
3	A-47	20-24	Upper platform, ford.	S.	1,730	6.4
5	A-47	20-24	Upper platform, ford.	S.	1,280	4.8
7	A-47	20-24	Upper platform, ford.	S.	1,590	5.9
2	A-46	20-24	Upper platform, ford.	P.	1,240	4.6
4	A-46	20-24	Upper platform, ford.	P.	1,630	6.1
6	A-46	20-24	Upper platform, ford.	P.	1,280	4.8
8	A-46	20-24	Upper platform, ford.	P.	1,590	5.9
	Total				16,040	59.7

Capacities of Reserve Feed-Water Tanks.

B-94	58½-62	Inner bottom, amid.	P.	5,696	21.2
B-95	58½-62	Inner bottom, amid.	S.	5,696	21.2
B-96	62 -65	Inner bottom, amid.	P.	4,863	18.1
B-97	62 -65	Inner bottom, amid.	S.	4,863	18.1
B-98	65 -67½	Inner bottom, amid.	P.	4,057	15.1
B-99	65 -67½	Inner bottom, amid.	S.	4,057	15.1
Total				29,232	108.8

Capacities of Feed and Filter Tanks.

C-1	80-83	Engine room, aft	S.	5,475	20.4
C-2	80-83	Engine room, aft	P.	5,475	20.4
Total				10,950	40.8

Capacities of Trimming Tanks.

A-1	Stern-3		S. & P.	4,475	16.6
A-2	3-5		S. & P.	7,467	27.7
D-12	101-104		S. & P.	14,853	55.1
D-13	104-Stern		S. & P.	10,162	37.7
Total				36,957	137.1

Recapitulation.

Capacity of fresh-water tanks	16,040	59.7
reserve feed-water tanks	29,232	108.8
feed and filter tanks	10,950	40.8
trimming tanks	36,957	137.1
Total, fresh water in tanks	93,179	364.4

Capacities of Double-Bottom Compartments.

Compartment.	Frames.	Side.	F.W.	S.W.
			Tons.	*Tons.*
A-93	12 to 15	P. and S.	33.2	34.2
A-94	15 to 18	P. and S.	39.7	40.8
A-95	18 to 21	P. and S.	47.8	49.2
A-96	21 to 24	P. and S.	51.4	52.9
A-97	24 to 27	P. and S.	54.5	56.1
A-98	27 to 30½	P. and S.	66.8	68.7
A-99	30½ to 35	P. and S.	90.4	93.0
B-80	35 to 38	P. and S.	71.2	73.2
B-81	38 to 40½	P. and S.	60.6	62.3
B-82	40½ to 44	P. and S.	87.1	89.6
B-83	44 to 47	P. and S.	75.5	77.7
B-84	47 to 49½	P. and S.	63.9	65.7
B-85	49½ to 53	P. and S.	89.8	92.4
B-86	53 to 56	P. and S.	76.6	78.8
B-87	56 to 58½	P. and S.	63.7	65.5
B-88	58½ to 62	Port	22.7	23.3
B-89	58½ to 62	Starboard	22.7	23.3
B-90	62 to 65	Port	19.2	19.8
B-91	62 to 65	Starboard	19.2	19.8
B-92	65 to 67½	Port	15.9	16.4
B-93	65 to 67½	Starboard	15.9	16.4
C-95	67½ to 71	P. and S.	88.3	90.8
C-96	71 to 74	P. and S.	74.3	76.4
C-97	74 to 77	P. and S.	73.0	75.1
C-98	77 to 80	P. and S.	71.9	74.0
C-99	80 to 83	P. and S.	70.0	72.0
D-96	83 to 86½	P. and S.	68.3	70.3
D-97	86½ to 90	P. and S.	64.7	66.6
D-98	90 to 94	P. and S.	67.7	69.6
D-99	94 to 98	P. and S.	68.2	70.2
Total			1,734.2	1,784.1

DESCRIPTION OF DECKS, BRIDGES, ETC.

Masts.—There are two steel masts, the foremast being located at frame 36 and the mainmast at frame 76. They carry the wireless telegraphy system, and each mast has a spar topmast, signal yard, and the following platforms: two on the foremast and three on the mainmast; the upper ones being for searchlights, the lower on the foremast and the middle on the mainmast being range-finder platforms, and the lower on the mainmast carrying searchlights. The conning tower and its support is the foundation for the foremast.

Flying Bridge.—This bridge, built around the foremast, above the forward half of the bridge, is of semi-triangular shape for about fifty feet of the ship's width, with a walk to

the sides of the ship. That portion forward of the mast is the pilot house, in which are the steering wheel, helm and revolution indicators, engine telegraphs, whistle and siren pulls, voice tubes to every part of the ship, etc. Two searchlights are carried, located port and starboard, about 30 feet from the center line of the ship.

Forward Bridge.—This bridge is of irregular shape, being like a rectangle with the corners cut off, is about fifty feet wide by 32 feet long, and is located above the forward end of the upper deck. The conning tower, with entrance to the same, are on this level, and a triangularly-shaped extension forward of the conning tower carries an armored torpedo-directing hatch and station platform. Aft of the conning tower is a chart house and emergency cabin for the commanding officer, the two rooms forming one structure, which is built entirely of brass, the roof of same being used as a platform for the compass. Two 3-pounder and two 3-inch R.F. guns are carried on this bridge.

After Bridge.—This bridge is built around the aftermast and just above the after end of the upper deck. It is about 22 feet long by 50 feet wide, with a walk to the sides of the ship. On it are located six 3-pounder guns. Aft of the mast and flush with the deck of the bridge is an armored torpedo-direction station that extends to the upper deck, the circular foundation for which, between the upper and main decks, being used as the wireless-telegraph station.

Upper Deck.—This deck extends from frame 35 forward to frame 80 aft. All boats are stored on this deck when at sea, except two 30-foot cutters, two 30-foot whale boats, one 30-foot gig whale boat and one 30-foot racing cutter. Two electrically-operated boat cranes, one port and one starboard, are located at frame 64 with their heels stepped on the gun deck; and forward of each, at frame 52, is an electric deck winch. Within the superstructure, under the bridges, hammock berthing has been provided and fresh-water tanks and vegetable lockers located, the space between the bridges being taken up with the general stowage of boats and terminus of exhaust

and ventilator trunks, and fireroom and engine-room hatches. Six 3-inch rapid-fire, four 3-pounder and two 1-pounder guns are located on this deck, as noted in Secondary Battery section of this article. The following boats are carried on this deck: one 40-foot and two 36-foot steam cutters; one 36-foot and three 33-foot launches; three 30-foot cutters, one 30-foot barge; two 20-foot, one 16-foot and one 14-foot dingies; two 14-foot punts.

Total number of boats carried on ship, ... 23
Total number of men carried in boats exclusive of the punts and racing cutter, ... 807

Main Deck.—The main deck is continuous, extending the full length of the ship. On it are six elliptical, electrically-operated turrets, two of them being 12-inch and four 8-inch, all mounting two guns each. The 12-inch turrets are on the center line of the ship, the center of barbette for the forward one being at frame 28¾, and that of the after one at 85½. Between these turrets, port and starboard, at frames 44½ and 69, are the 8-inch turrets, the centers of the barbettes being 28 feet from the center line of the ship, and the superstructure which extends from frames 35 to 79½ cut away so as to give the forward guns a range of 90 degrees forward and the after guns the same range aft. Within the superstructure, between the 8-inch turrets, six 3-inch R.F. guns are located, three being on the port and three on the starboard side. The arrangement on this deck, beginning forward, is as follows: four deck openings for hawse pipes between frames 6 and 9; two anchor davits, one port and one starboard, at frame 9; two 20-foot dingies, harbor position on davits, one port and one starboard, between frames 14 and 18; covered superstructure over anchor-windlass pocket between frames 22 and 24½, the chains passing out through chain pipes in the forward side; two electrically-operated deck winches for anchor work, coaling and general use, one port and one starboard, between frames 25 and 26; two 30-foot cutters, harbor position on davits, sea position on deck, one port and one starboard, be-

tween frames 30 and 36. Aft of the forward bulkhead of the superstructure, at frame 35 to 40, and about 34 feet in width, is the crew's galley; galley issuing room, frames 40 to 41½; No. 1 forward fireroom-exhaust hatch, frames 42½ to 43½; No. 1 boiler hatch, frames 43½ to 46½; general mess pantry, frames 46½ to 48½; No. 1 after fireroom-exhaust hatch, frames 48½ to 50½, with coaling trunks, port and starboard; No. 2 boiler hatch, between frames 52 and 56; officers' galley, frames 56 and 58½; No. 2 fireroom-exhaust hatch, frames 58½ to 59½; No. 3 boiler-hatch, between frames 61 and 65; No. 3 fireroom-exhaust hatch, frames 65 to 66½, with coaling trunks, port and starboard; hammock room, frames 66½ to 70; paymaster's office, starboard, executive officer's office, port, frames 72 to 74; engine hatch, frames 74 to 78; wireless-telegraph room on the center line of the ship, and battalion lockers, port and starboard, frames 78 to 79½, the after bulkhead of the superstructure being located at the latter frame. Aft of the superstructure there are two 30-foot whale boats, one port and one starboard, on davits between frames 78 and 84; two electrically-operated deck winches at frame 90; one 30-foot gig whale boat, starboard, and one 31-foot racing cutter, port, on davits between frames 100 and 105. Outside the superstructure the deck is covered with 3½ by 3½-inch Virginia pine, and inside with linoleum.

Gun Deck.—This deck is also continuous, and the full width of the ship. It has within its space the barbettes for the 12-inch and 8-inch turrets, besides which, in splinter-proof compartments, in sponsons, there are twelve 7-inch R.F. guns. Beginning forward, the arrangement is as follows: Stern to frame No. 8, crew's showers; frame 8 to 14½, crew's wash room; frame 14½ to 28, crew's quarters, with windlass engine amidships, with quarters for master-at-arms on the starboard, and sergeant of marines on the port. Between the 7-inch athwartships armor, at frames No. 28 and No. 86, and between the 12-inch barbettes, the space is divided into twelve splinter-proof compartments, by 1½-inch nickel-steel plate, each compartment containing one 7-inch R.F. gun and all

necessary ammunition hoists and signaling devices. Amidships, between frames 36 and 39, are the bakery, starboard, and bread-mixing rooms, port; No. 1 boiler hatch between frames 41½ and 47, containing drying rooms, entrance hatches, boiler uptakes and ventilator ducts; between frames 46 and 50 is the printing office, starboard, and prison, port; frames 49 to 51 amidships are drying rooms and entrance hatches to middle firerooms; between frames 52 to 56 is No. 2 boiler hatch; between frames 57½ and 60 are drying rooms and entrance hatches to the after firerooms; between frames 61 and 66½ is No. 3 boiler hatch, containing cleaning and drying rooms, entrance hatches to the after firerooms, uptakes and ventilator ducts; amidships, between frames 68 and 78, are the ward-room officers' lavatory and bath, armory and machinery hatch, starboard, and junior officers' lavatory and bath, engineer's office and machinery hatch, port; and outboard, between frames 64 and 68, are the closets for the ward-room officers, starboard, and junior and warrant officers, port; amidships, between frames 79½ and 82, are the ordnance officer's office, starboard, and navigator's office, port. Aft of the 7-inch athwartship armor, at frame 82 to frame 107, the arrangement is as follows: Starboard—Staterooms for the navigator, ordnance and watch officers, captain's bath, stateroom cabin, spare stateroom and bath. Port—Staterooms for senior engineer, senior medical, marine and executive officers, executive officer's bath, captain's office, ward-room mess room. Aft of frame 107 to stern of ship is the captain's after cabin.

Berth Deck.—The arrangement of this deck, beginning forward, is as follows: Frames 3 to 5, oils; frames 5 to 9, starboard—lamps and oils, port—paints; frames 9 to 12, starboard—equipment stores, port—paints; frames 12 to 16, amidships—blower room, starboard—isolation ward and sick-bay lavatory, port—commissary stores and band room; frames 16 to 24, sick bay, containing the operating and dispensary rooms; frames 24 to 35, ice-machine and refrigerator rooms, starboard, and machinists' and petty officers' quarters, port; frames 35 to 40½, starboard—laundry, port—general mess

stores and issuing room; frames 40½ to 72, outboard, starboard and port, are six coal bunkers, inboard of which is a continuous passage, and amidships are the firemen's and servants' wash rooms and the evaporator room; frames 72 to 78, starboard and port, is crew space; and amidships, between frames 73½ and 82, is the general workshop; frames 78 to 94 has junior officers' staterooms and mess rooms, starboard, with the same for the warrant officers, port; frames 94 to 98, junior officers' quarters; frames 98 to 108 has five staterooms for ward-room officers on each side of the ship; frame 108 to stern, ward-room officers' showers.

Protective Deck.—The space under the berth deck on the slope at the ship's sides is divided into numerous compartments and used for various stores, excepting the space between frames 41 and 72, which is a continuation of the berth-deck coal bunkers.

Upper Platform Deck.—The arrangement, beginning forward, is as follows: Stern to frame 5, trimming tank; frames 5 to 9, ordnance stores; frames 9 to 12, starboard—marines' stores, port—medical stores; frames 12 to 16, starboard—dry provisions, port—clothing and small stores; frames 16 to 20, starboard—dry provisions, port—paymaster's issuing room and dry provisions; frames 20 to 24, fresh-water tanks; frames 24 to 35, amidships—four 12-inch shell rooms and 12-inch handling room, starboard—two 12-inch and two 8-inch powder magazines and one 8-inch shell room; port side—similar; frames 35 to 40½, amidships—forward dynamo room; frames 40½ to 67½, divided into six boiler compartments; frames 35 to 46, starboard and port—8-inch ammunition passage and coal; frames 46 to 67½, starboard and port—coal; frames 67½ to 72, amidships—after dynamo room; frames 72 to 83, engine rooms; frames 67½ to 83, starboard and port—8-inch ammunition passage and coal; frames 83 to 94, the arrangement of handling room and magazines is similar to that noted for the forward end of the ship; frames 94 to 98, starboard—navigator, ward room and officers' stores, port—captain and warrant officers' stores; frames 98 to 104, amid-

ships—steering-engine room, starboard and port—electric stores; frames 104 to stern, steering gear.

Lower Platform Deck.—The arrangement, beginning forward, is as follows: Frames 12 to 16, starboard—awnings and linoleums, port—construction stores; frames 16 to 21, torpedo room; frames 21 to 24, amidships—chain lockers, starboard and port—upper hold; frames 24 to 35, amidships—7-inch powder magazines, starboard—7-inch powder magazines, 7-inch shell, 3-inch powder magazine, port—7-inch powder magazine, 7-inch shell, 3-inch ammunition; frames 35 to 40½, amidships—coal; frames 40½ to 67½, amidships—boiler compartments; frames 67½ to 72, coal; frames 72 to 83, engine rooms; frames 40½ to 83, outboard of machinery spaces, ammunition passage, with ten coal bunkers outboard of these on each side of the ship; frames 83 to 84, ammunition passage; frames 84 to 94, amidships—7-inch powder magazines; frames 84 to 94, starboard and port—7-inch shell, 7-inch powder and 3-inch ammunition; frames 94 to 98, torpedo room.

Hold.—The arrangement, beginning forward, is as follows: Stern to frame 5, trimming tank; frames 5 to 18, divided into four compartments on each side of the ship for dry provisions; frames 18 to 21, starboard—naval-defence stores, port—torpedo gear and mine and war heads; frames 21 to 24, amidships—chain lockers, starboard and port, lower hold; frames 24 to 35, amidships—3-inch ammunition magazine, starboard—3-inch powder and saluting powder, 1-pounder and 3-pounder magazines, port—spare magazine, 3-inch field ammunition, and small arms magazines; frames 35 to 67½, boiler spaces and coal, outboard; frames 67½ to 72, coal; frames 72 to 83, engine rooms and coal, outboard; frames 83 to 94, amidships—3-inch ammunition, outboard, starboard and port—machinery stores; frames 94 to 98, starboard—ordnance stores and air compressors, port—war heads and ordnance stores; frames 98 to 101, fresh water; frames 101 to stern, two trimming tanks.

MAIN ENGINES.

There are two main engines, right and left-hand, outboard turning when going ahead, each in a water tight compartment. They are of the vertical, inverted, direct-acting, 4-cylinder, triple-expansion type, designed to develop about 16,500 I.H.P. at about 120 R.P.M., with a steam pressure of 250 pounds at the H.P. cylinder. The arrangement of cylinders, beginning forward, is forward L.P., H.P. and I.P., and after L.P. respectively; the F.L.P. and H.P. and A.L.P. and I.P. being bolted together, thus allowing freedom for expansion between the H.P. and I.P. cylinders in fore-and-aft and vertical planes, while a system of tie rods and braces prevent fore-and-aft and athwartship motion to the engine as a body. The material used for the cylinders is cast iron, fitted with close-grained, hard cast-iron piston and valve-chest liners, the $\frac{3}{4}$-inch space between the casings and piston liners being utilized as the steam jacket, the top and bottom heads of the I.P. and L.P. cylinders also being jacketed. Steam for the jackets is taken from the boiler side of the main-engine throttle valve, and passes successively through the cylinders, spring-reducing valves at the I.P. and L.P. cylinders, allowing a maximum steam pressure of 80 and 60 pounds respectively. Each end of the piston valves is fitted with a packing ring of hard cast iron, made practically solid, being turned larger than the bore of the valve-chest liners, cut obliquely, the abutting ends bolted together and finished to fit the liners. The body of the H.P. valve is cast iron, and those of the I.P. and L.P. valves, steel pipe with cast-iron heads. The H.P. valve stem is connected to the link block, and the I.P. and L.P. stems to a cast-steel crosshead which is connected to the link block, the Stephenson double-bar link motion, with adjustable cut-off, being used. There are five graduations for points of cut-off on each reverse-shaft arm, and are for the ahead motion only.

The cylinders are supported by a framing consisting of twelve forged-steel columns, $5\frac{1}{2}$ inches diameter, flanged top and bottom for bolting to the cylinders and bed plate; each athwartship pair of columns being trussed by X braces, and

stiffened fore and aft by horizontal and diagonal tie rods leading from the middle pairs of columns, thus offering no resistence to cylinder expansion.

The crosshead guides, which are of cast iron, are bolted at the top to facings provided on the cylinders, and at the bottom to cast-steel I-section strongbacks carried by the engine framing. The guides, which are of the slipper type, are hollow for the circulation of water to keep them cool.

The bed plates are of cast steel, made in three sections, flanged and securely bolted together by body-bound bolts. Each bed plate when assembled consists of two longitudinal and six cross girders of I-section, well stiffened by ribs, and furnishes the seatings for the crank-shaft bearings, engine framing and turning engine. Body-bound bolts secure the bed plates to the keelsons, forged-steel washers being fitted at each bolt between the bed plate and the keelson, the space between the washers being filled in with yellow pine.

The crank shaft is arranged with the cranks 90 degrees apart and is in two sections, the forward L.P. and H.P. being in one, and the I.P. and after L.P. on the other, the sequence of cranks being H.P., I.P., F.L.P. and A.L.P.

The upper ends of the connecting rods are forked for the crosshead-pin brasses; these brasses and those of the crank pin being lined with white metal. Crank shafts and connecting rods are high-grade machinery forgings. The pistons are conical shaped, rough machined all over, the material of the H.P. being cast iron, and that of the I.P. and L.P. cast steel, the followers in each case being of the same material. The pistons are bored taper to fit the rods, and have a square counterbore on the under side to suit the shoulder on the piston rods. Pistons are secured to the rods by steel nuts, locked in place by plates bolted to the piston.

The packing rings are of hard cast iron, being practically solid, having been cut obliquely for machining and the ends clamped solidly together; lugs cast on the back of the rings limit the play, the bore being $\frac{1}{8}$ inch greater than the corresponding bore of the piston.

The piston rods are high grade, and the crossheads and slippers class "A" forged steel, the slippers being faced with white metal. The ends of the rods are taperedto fit the pistons and crossheads, and the slippers are bolted to the crossheads.

Engine Framing.—The reversing engine is located inboard and is bolted to the top of the H.P. cylinder. There is an oil-control cylinder of brass connected to the steam cylinder by wrought-steel stanchions which serve as crosshead guides. The reverse shaft is also on the inboard side of the engines, near the top of the engine framing, and is connected to the crosshead of the reversing engine by two forged-steel connecting rods. Differential levers control the valve motion.

The turning engine is located inboard on the after end of the main engine-bed plate. The shaft carries a worm, that engages a worm wheel carried by an inclined shaft, at the lower end of which is another worm that engages the driving worm wheel located on the coupling flanges between the crank and thrust shafts. The second worm is keyed to the inclined shaft by means of a feather, but may be moved vertically to engage or disengage the worm wheel on the crank shaft. A square for a ratchet, for turning by hand, has been provided on the after end of the turning-engine shaft.

MAIN ENGINE DATA.

CYLINDER DATA.

	H.P.	I.P.	L.P.
Diameter, inches	32 1/8	53	61
Stroke, inches	48	48	48
Thickness of body, inches	1 7/8	1 7/16	1 5/16
Thickness of liner, inches	1 1/8	1 5/16	1 1/2
jacket space, inch	3/4	3/4	3/4
valve chest, inches	1 7/16	1 1/2	1 1/8
Number of studs in cylinder covers	36	48	56
Diameter of studs in cylinder covers, inches	1 1/4	1 3/8	1 1/2
Pitch of studs in cylinder covers, inches	4 5/8	4 5/8	4
Number of studs in valve-chest covers	18	18	16
Diameter of studs in valve-chest covers, inch	1	1	1
Pitch of studs in valve-chest covers, inches	4 3/4	4 5/8	5 1/4
Number of tap bolts in piston liners	24	36	36
Diameter of tap bolts in piston liners, inch	7/8	7/8	7/8
Pitch of tap bolts in piston liners, inch	4 1/4	4 1/8	5 1/4

	H.P.	I.P.	L.P.
Number of piston valves	1	2	2
Diameter of piston valves, top	17½	19	19
bottom	17 7/16	18 15/16	18 15/16
balance pistons, inches	9	9	9
Port area through valve-chest liner, top, inches	150.5	166.92	190.83
bottom, inches	158.7	166.22	190.04
Valve travel, inches	10	10	10

VALVE DATA FOR PORT ENGINE.

	H.P.	I.P.	L.P.
Steam lead, top, linear, inches	⅛	⅝	1 1/16
bottom, linear, inches	9/16	15/16	1⅛
Full gear cut-off, top, per cent	80.8	76.5	74.0
bottom, per cent	72.2	69.0	63.7
Earliest cut-off, top, per cent	56.0	51.5	51.7
bottom, per cent	44.5	43.7	39.9
Diameter of valve stem through gland, inches	2⅞	2⅞	2⅞
valve, inches	1⅞	1⅞	1⅞

	H.P. and I.P.		L P.
Diameter of piston rods, inches	7¼		7¼
axial hole, inches	1½	3	4⅝
crosshead pins, inches	9½		9½
Length of crosshead pins, each, inches	9½		9½
slippers, inches	27½		27½
Width of crosshead slippers, inches	21		21
Length of backing surface, inches	27½		27½
Width of backing surface, each, inches	7¾		7¾
Length of connecting rod, center to center, inches	96		96
Diameter of connecting rod, top, inches	7		7
bottom, inches	8		8
hole, body, inches	1½	3⅝	4⅞
crosshead end, inches	1½	1½	1½
pin bolts, inches	3		3
Number of crosshead-pin bolts	4		4
Diameter of crank-pin bolts, inches	4		4
Number of crank-pin bolts	2		2
Diameter of crank pins, inches	17½		17½
Length of crank pins, inches	19		19
Diameter of hole in crank pins, inches	10		10
shaft, inches		9½	
crank shaft, inches		17½	
journals, inches		16½	
coupling flanges, inches		29½	
Width of crank-shaft coupling flanges, inches		3¼	
Number of coupling bolts		16	
Diameter of coupling bolts, inches		3¼	
pitch circle, inches		23½	

Width of crank webs, inches	19
Thickness of crank webs, inches	10½
Number of main bearings	6
Length of main bearings, H.P., inches	19¼
I.P., inches	19½ and 10¾
forward L.P., inches	19½
after L.P., inches	10¾ and 19½
Diameter of main-bearing bolts, inches	2⅝
Number of main-bearing bolts per bearing	4
Number of packing rings in pistons	1
Width of packing rings in pistons, inches	2½
Thickness of packing rings in pistons, inch	1
Number of follower studs in H.P. pistons	10
I.P. pistons	16
L.P. pistons	17
Diameter of follower studs, inches	1¼

REVERSING ENGINE.

Diameter of steam cylinder, inches	13
oil cylinder, inches	7
steam piston rod, inches	2⅞
oil piston rod, inches	2⅞
Stroke, inches	19⅝
Length of connecting rods, feet and inches	3-05 $\frac{13}{16}$
reverse levers, inches	16⅛
Diameter of reverse shaft, inches	7 and 6
bearings, inches	6½
Length of reverse-shaft bearings, inches	8¼

TURNING ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	6
piston rods, inches	1½
valves, inches	3½
Stroke, inches	7
Valve travel, inches	1½
Crank angle, degrees	90
Area of crosshead slipper, inches	16⅛
Diameter of crosshead pin, inches	1¾
Length of crosshead pin, inches	2⅝
Diameter of crank pin, inches	2½
Length of crank pin, inches	2½
Diameter of crank shaft, inches	2½
Length of crank-shaft bearings, inches	4 and 3
Diameter of pitch circle of worm, inches	6 $\frac{13}{16}$
wheel, inches	17 $\frac{3}{16}$
Number of teeth	36

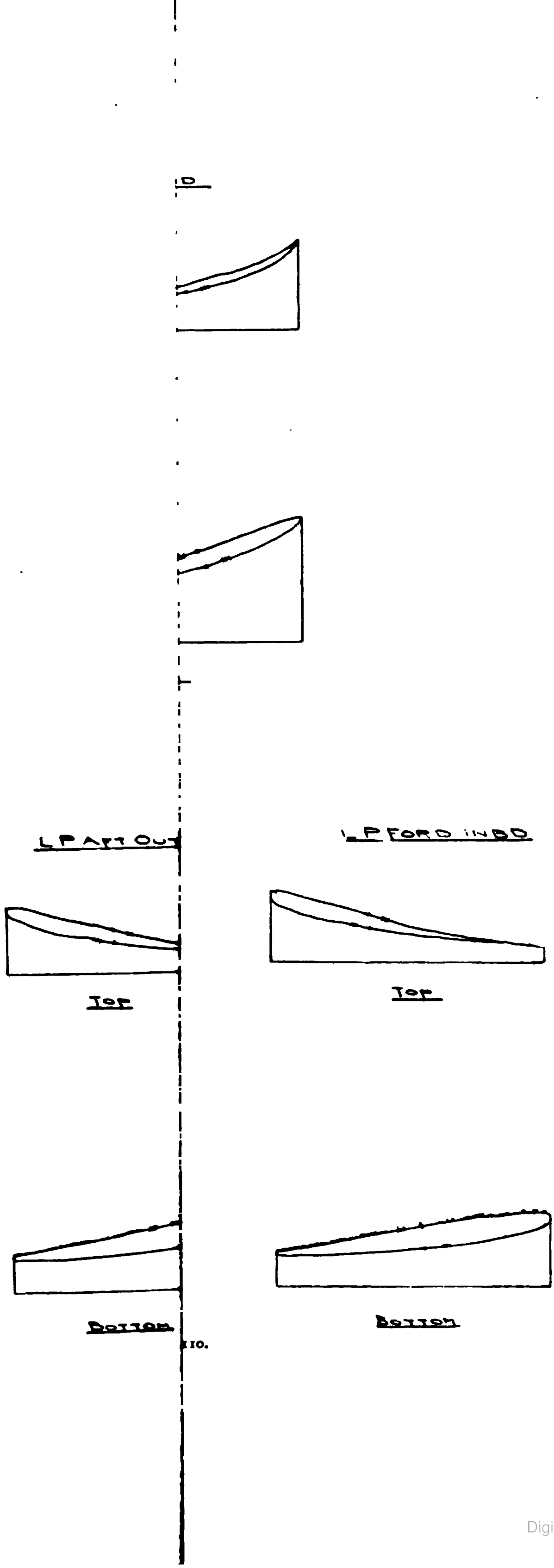
Diameter of pitch circle of vertical worm, inches	9
worm wheel, inches	$64\frac{15}{32}$
Number of teeth	60
Diameter of vertical shaft, inches	$4\frac{1}{4}$
Area of step bearing, square inches	$38\frac{1}{4}$

ASSISTANT CYLINDERS.

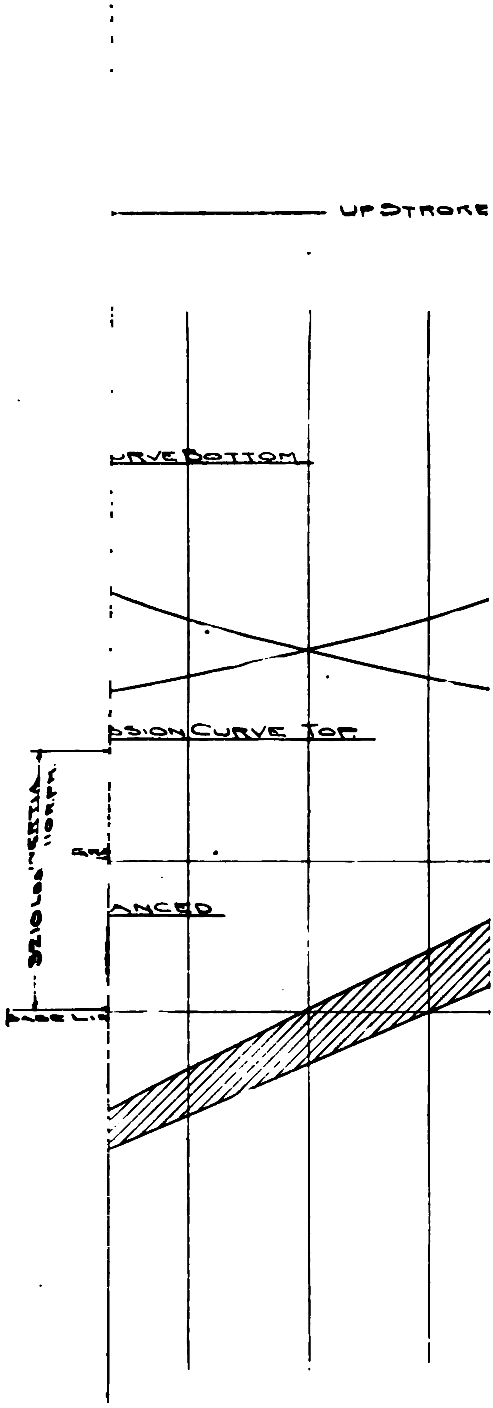
An interesting and important feature of the propelling machinery is the Lovekin Improved Assistant Cylinders, applied to the valve gear of the main engines, this ship being the second in the United States Navy to be equipped with this simple and yet most effective device for overcoming the inertia primarily the cause of considerable wear upon the heavy valve gear working at high speeds with the resulting item of expense for repairs; and an unnecessary amount of vibration so annoying at any time and especially so on shipboard.

The highly satisfactory results obtained were anticipated, as the cylinders had proven their usefulness upon the armored cruiser *Washington*, and were the subject of an exhaustive article at that time; but an elaboration of diagrams and calculations were not necessary to prove to the practical man that the device was effective, as the smooth-running valve gear was sufficient evidence of that.

For results obtained upon the *Kansas*, there is given here (Plates III and IV) an analytical diagram that briefly illustrates the work done by the assistant cylinder on the port H.P. valve gear. The ordinates of the lines A B and B C, measured from the base line, give the inertia resistance to be overcome by the cylinder at the various stages of the valve travel. The compression and expansion curves are exact reproductions of cards taken on the trial. The difference in the lengths of the ordinates of the upper and lower curves gives the ordinate of the resulting curve; and the shaded area between the inertia lines and the resultant effort curve represent the unbalanced work which has to be done by the eccentrics and is fully explained in the following articles, the



LP AFT OUT
LP FORD INBD
TOP
TOP
BOTTOM
BOTTOM
110.



ls.

limited scope of this article not permitting an exhaustive analysis of the subject. See Plates Nos. III and IV.

Interested readers will be well repaid by consulting the following references: "Balance Cylinders, Theoretically and Practically Considered," "Performance of the Assistant Cylinders of the *Washington*," and JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XVI, No. 3; XVII, No. 3; XVIII, No. 3, and "The Balancing of Valve Gears," "Journal of the Society of Naval Architects and Marine Engineers," Nov., 1902.

SHAFTING AND THRUST BEARING.

The shafting is arranged in four sections, as follows: Thrust, line, stern-tube and propeller, all of class A, forged steel. The line and stern-tube shafts are coupled as follows: A forged-steel collar fits on the end of the stern-tube shaft, secured, to prevent turning, by three keys, and into a circumferential groove in the end of the shaft is fitted a ring made in halves, the coupling bolts passing through the steel collar, the half rings and the flange on the thrust shaft. The stern tube and propeller shafts are coupled as follows: Over the tapered ends of the shafts is fitted a long sleeve bored with a double taper to fit that of shafts and secured to each by two keys and a taper cotter.

The thrust bearing is of the horseshoe type, with a steady bearing at each end, fitted with glands to prevent the escape of oil. The body is of cast iron, so shaped as to form an oil reservoir, and the horseshoes are of cast steel, made hollow for the oil and water service. They are faced with white metal and have oil grooves on the bearing surface.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	15¼ and 15⅝
hole, inches	9½
Number of thrust-shaft collars	12
Outside diameter of thrust-shaft collars, inches	24½
Width of thrust-shaft collars, inches	2
space of thrust-shaft collars, inches	4
Length of thrust-shaft bearings, inches	18

Number of thrust shoes	12
Diameter of thrust side rods, inches	3¾
line shafting, inches	15⅛
stern-tube shafting, inches	16½
shaft hole, inches	9⅛
Length of stern-tube shaft forward bearing, inches	54
after bearing, inches	54¼
Diameter of propeller shaft, inches	16½
hole, inches	9⅛
Length of strut bearing, inches	51
Diameter of thrust-shaft coupling flange, inches	29½
Thickness of thrust-shaft coupling flange, inches	3¼
Number of thrust-shaft coupling bolts	8
Diameter of thrust-shaft coupling bolts, inches	3
pitch circle, inches	23⅛
Length of collar on stern-tube shaft, inches	11¾
Diameter of collar on stern-tube shaft (inside), inches	16⅛
Number of keys	4
Length of keys, inches	11⅝
Width of keys, inches	2¼
Depth of keys, inches	1⅝
Thickness of halved ring, inches	2
Inside diameter of halved ring, inches	14⅛
Length of outboard coupling sleeve, inches	67¼
Outside diameter of outboard coupling sleeve, inches	21¾
Width of longitudinal keys, inches	2¼
Depth of longitudinal keys, inches	1⅛
Width of cotter keys, top, inches	7⅛
bottom, inches	6⅛
Thickness of cotter keys, inches	2

PROPELLERS.

The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a composition nut on the end of the shaft, the nut being threaded the reverse of the direction of the propeller. The nut is locked in place and covered with a composition cap bolted watertight to the hub. The blades fit into tapered recesses in the hub, being held in place by rolled manganese tap bolts, composition chocks being fitted around the bodies of the bolts to keep the blades from shifting after being set to the desired pitch. The bolt holes in the hub are oval in shape and allow a pitch adjustment of one foot either way. The pro-

pellers were accurately balanced, and each blade was made a true surface on both faces, by machining, grinding and polishing. The material for blades and hubs is manganese-bronze.

DATA FOR ONE PROPELLER.

Number of blades	3
Diameter of propeller, feet and inches	17-04
Pitch for official trial, feet and inches	17-06
maximum, feet and inches	18-06
minimum, feet and inches	16-06
Ratio of pitch to diameter	1.009
Helicoidal area, square feet	96.0
Projected area, square feet	82.5
Disc are, square feet	235.97
Ratio projected to disc area	.35
Area of immersed midship section, square feet	1,811.0
Ratio disc to immersed midship-section area	.26
projected to immersed midship-section area	.091
Number of tap bolts for blades	27
Diameter of tap bolts for blades, inches	3⅛
Length of tap bolts for blades under head, inches	6½
Width of longitudinal key, inches	3½
Thickness of longitudinal key, inches	2¼
Length of longitudinal key, inches	36
Diameter of shaft-nut thread, inches	9½
Length of shaft-nut thread, inches	9
Number of threads per inch	4
Immersion of tip of blade, inches	74⅞
Tip of blade above keel, inches	10⅝

DATA FROM OFFICIAL TRIAL.

	Port.	*Starboard.*
Average revolutions per minute on course	120.52	122.12
Slip, per cent. of its own speed	14.29	15.43

BOILERS.

The boilers on this ship are of the Babcock & Wilcox type, and were designed, built and installed by this well known concern. The battery consists of sixteen boilers placed two each in eight watertight compartments, all compartments being symmetrically arranged.

Each boiler has one drum extending the full width of the boiler and is located within the casings, so that about one-

fourth of the shell is in contact with the gases at the top of the combustion chamber. Immediately below the drum are the front headers, and at the back of the boilers are the back headers, connected by tubes set at an angle of 15 degrees.

The weight of the drum and front headers, etc., is carried by the furnace front, and that of the back headers by a light structural-steel girder, that also forms a portion of the casing.

This is protected from the fire by a bridge wall of brick extending the full width of the furnaces. There are two of these of equal size, separated by a wall of rectangular water legs that extends from the lower row of tubes to a point just below the surface of the grate. The side walls are similarly constructed, and all are connected to a mud drum carried in the boiler front immediately below and connected to the front headers, thereby providing a constant circulation of water and affording excellent means for draining and blowing off.

The boiler casings are made of steel plates and shapes, fire brick and block magnesia being used for insulation. Front and back at the header space they are in the form of doors of such size that they may be easily removed for access to the inside of the tubes, and the sides are removable in sections for access to the baffle plates and for cleaning soot from the tubes.

Each furnace is provided with three furnace and two ash-pit doors, and the grate bars are of the ordinary common type, of such size and shape as to be easily handled by one man.

Boiler Drums.—The boiler drums are of open-hearth steel plate, having two seams which are butt-strapped inside and out. The heads are convex, the radius of which is equal to the inside diameter of the drums, and in each is a flanged manhole shaped from the same plate. All butt straps were formed to the proper curvature of the drum under hydraulic pressure, the rivet holes being punched about ¼ inch smaller than the diameter of the rivets used, and then drilled out full size after the plates were rolled and assembled. After drill-

ing all burrs were cleaned off, the plates reassembled, with turned bolts holding the parts in place for riveting. Wrought-steel pads securely riveted to the drum shell are provided for attaching all valves and fittings.

The following external fittings are provided for each drum:

One 5-inch hand- and steam-closing boiler stop valve.
One twin-spring safety valve, each valve 4½ inches in diameter.
Two 2½-inch stop valves for main and auxiliary feed.
Two 2½-inch check valves for main and auxiliary feed.
Two Klinger reflex water gauges with automatic fittings.
One 1½-inch surface-blow valve.
One ½-inch sentinel valve.
One ½-inch air valve.
One ¼-inch steam-gauge connection.
Three ⅝-inch try cocks.

The following internal fittings are provided:

One 6-inch brass dry pipe.
One 2½-inch main feed pipe.
One 2½-inch auxiliary feed pipe.
One 1½-inch pipe and scum pan for surface blow.
Zinc protectors and baskets.

Headers.—The headers are of forged steel, forged in one piece into a rectangular section, having male and female sinuous side for nesting when assembled. The front side is provided with numerous handholes for access to the generating tubes. The covers for these handholes are of pressed steel held in place by one stud, nut and dog. All holes for the generating tubes and connecting nipples between headers and drums are bored and reamed the size of tubing used.

Tubes.—The generating tubes are straight and made of seamless cold-drawn steel. These and the header and drum nipples, which are of the same material, are rolled in place by the B. & W. patent expander. Only one nipple showed any sign of leak on the trial.

DATA FOR BOILERS.

Number of boilers	12
Length of drum, feet and inches	15-03
Inside diameter of drum, inches	42
Thickness of drum plate, inches	$\frac{21}{32}$
Heating surface per header and tubes, square feet	175.8
Total heating surface for one boiler, square feet	4,396
grate surface for one boiler, square feet	91.56
H.S. for all boilers, square feet	52,752
G.S. for all boilers, square feet	1,000
Ratio H.S. to G.S	47.95
Number of boilers per smoke pipe	4
Height of smoke pipe above grate, feet and inches	92-05
Area through smoke pipe, square feet	53.46
Ratio G.S. to smoke-pipe area	6.85
Size of generating tubes, inches	2 and 4
Total number of headers, per boiler	25
Thickness of 2-inch generating tubes, B.W.G	8
4-inch generating tubes, B.W.G	25
Exposed length of generating tubes, feet and inches	9-0
Number of 2-inch generating tubes per boiler	786
4-inch generating tubes per boiler	25
Inclination of tubes, degrees	15
Height of boiler, feet and inches	13-06$\frac{1}{4}$
Length of boiler, feet and inches	10-01
Width of boiler, feet and inches	16-04$\frac{1}{8}$
Weight of one boiler complete, dry, pounds	66,791
water in one boiler, pounds	15,208
one boiler complete, wet, pounds	81,999
Total weight of boilers, dry, pounds	1,068,656
water in boilers, pounds	243,328
boilers, wet, pounds	1,311,984
Weight of boiler, dry, per cubic foot, pounds	29.89
wet, per cubic foot, pounds	36.70
Ratio of weight of boilers dry to boilers wet	.814

MAIN STEAM LINE.

The main steam pipes are arranged symmetrically in two systems, one on each side of the center-line bulkhead. They are united in the engine rooms by an 8-inch cross-connecting pipe, with a valve worked from both sides of the center-line bulkhead. The main steam pipes are supplied directly from the auxiliary steam pipes by a 7-inch globe valve at the after end of each boiler room, consequently there are no main steam lines in the forward boiler compartments. From the forward

and middle compartments the branches unite into a 9-inch pipe, which is increased to 11 inches at the junction of the connection from the after compartment. This diameter is carried aft through the dynamo rooms to the engine rooms, passing through the main-engine separators which are located on the center-line bulkhead in the forward end of each engine room, directly under the protective deck. The same diameter is carried from the separators to the main-engine throttle, and between these points a main stop or cut-out valve is located at the separator, and the expansion in this line is taken care of by a balanced expansion joint anchored to the forward engine-room bulkhead. From a 6-inch nozzle on the separators a cross connection is led between the main and auxiliary steam lines. At the separators nozzles are also provided for 5-inch branches to the receivers, and 5-inch bleeder connection to main condensers.

Globe stop or cut-out valves are placed just forward of each junction, and all valves in the main steam line have by-pass valves attached to relieve them when jammed on their seats.

The pipes are of seamless-drawn steel tubing with rolled-steel flanges. The pressure on the piping is 265 pounds per square inch, and the thicknesses of the pipes were determined by the following formula:

$$T = \frac{P \times D}{10{,}000} + \tfrac{1}{8} \text{ inch};$$

where

P = pressure above atmosphere, in pounds, per sq. inch.
D = inside diameter of pipe, in inches.
T = thickness, in inches.

The expansion of the main steam pipes in the firerooms and dynamo rooms is taken care of by an expansion joint in each compartment, and where pipe is bolted to athwartship bulkheads proper allowance has been made for expansion by making bolt holes oval.

Where condensed steam is apt to accumulate, drain valves

are fitted and pipes led to automatic float traps which discharge into condensers or feed tanks.

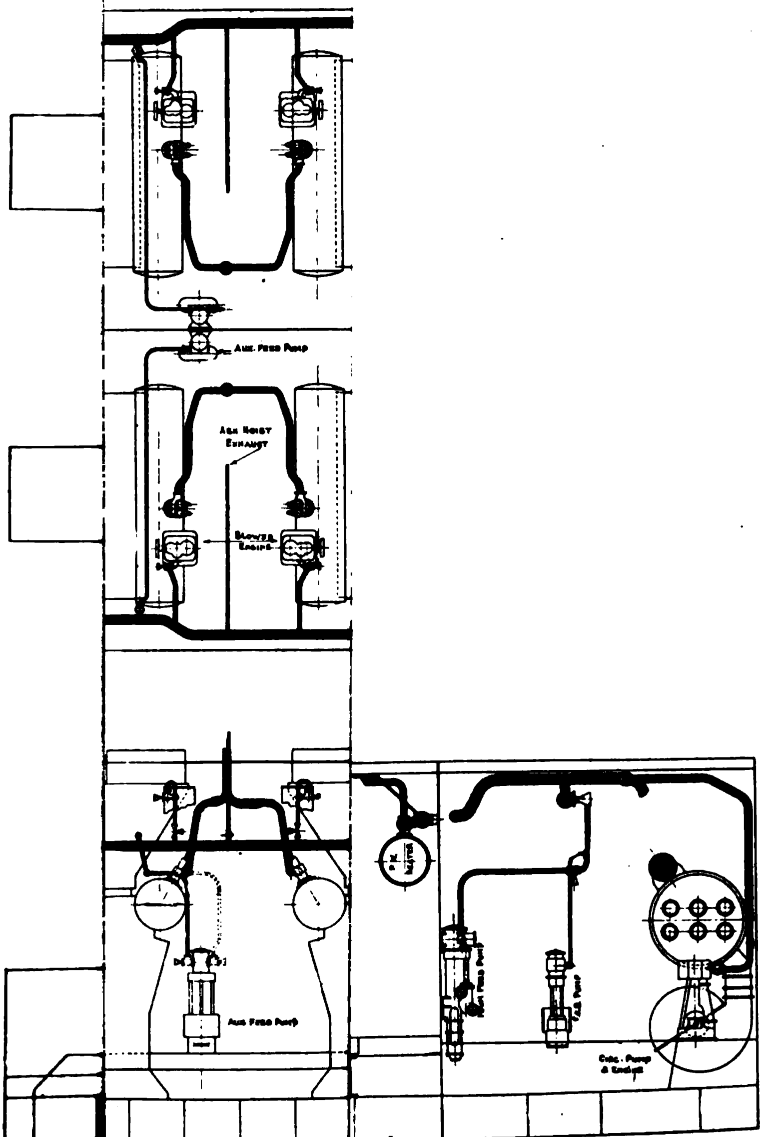
The piping is lagged with sectional magnesia covering with canvas sewed on and well painted and lagged with Russia sheet iron.

AUXILIARY STEAM LINES.

The auziliary steam lines are arranged in two systems, extending fore and aft through the boiler compartments, one on each side of the center-line bulkhead, directly under the main steam line. Each system is supplied directly from the boilers by connecting with the main stop valves of all boilers on its own side of ship through 5-inch branches. From the connection on the forward boiler in the forward boiler compartment the size of the pipe is carried 7 inches to the after end of the after boiler compartment, from which a 4-inch branch is taken on the port side to supply steam to the evaporating plant. Aft of this the pipe is reduced to 6½ inches diameter, and carried through the dynamo room to the engine rooms, with stop or cut-out valves on the forward dynamo-room bulkhead. Between these valves and the forward engine-room bulkhead are 4-inch branches for steam to after dynamo room, and aft of these branches the pipe is reduced to 6 inches diameter to supply steam to the auxiliaries in the engine rooms.

With a valve at the forward dynamo-room bulkhead the auxiliary steam lines are carried along the inboard sides of the engine rooms, and the two systems are united by a 6-inch connection passing through the center-line bulkhead with a valve at the bulkhead.

Steam to the steering engine is supplied by a 4½-inch branch taken off of the auxiliary steam line in the engine room on the starboard side. The distributing castings in the engine room also supply steam to main feed pumps, main air pumps, main circulating-pump engines, engine-room fire and bilge pumps, distiller circulating pumps, auxiliary air and circulating pumps, fresh-water pumps, oil pumps, turning and reversing engines, cylinder jackets, after heating system and



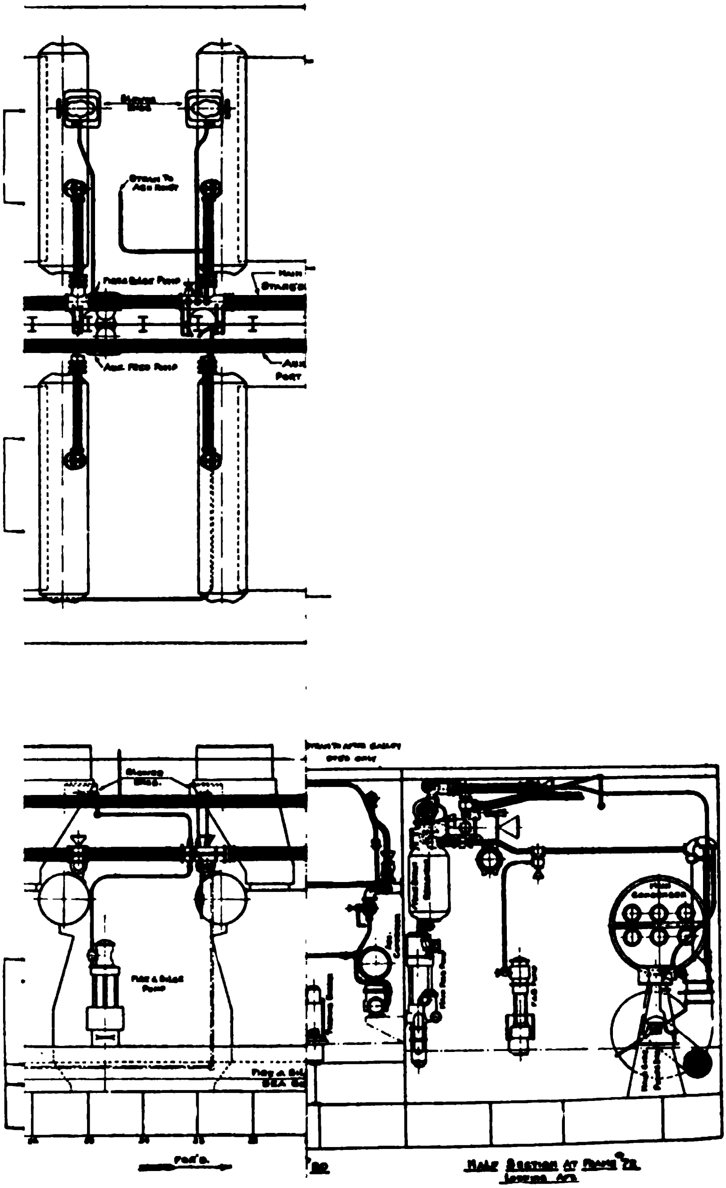
ON STB'D SIDE
IS INBOARD

#72

HALF SECTION AT FRAME #72
LOOKING AFT

KANSAS."——Plate V.

"KA



"KANSAS."—Plate VI.

galley, and connections for blowing out sea chests and boiling out condensers, feed-water heaters and grease extractors.

In the forward fireroom just forward of the 5-inch connection to the forward boiler the systems are reduced to 6 inches and fitted with cut-out valves. These 6-inch connections extend to the forward end of the compartment and pass through the center-line bulkhead, with a valve worked from both boiler rooms, thus forming a loop. At this forward cross connection are nozzles for 4-inch steam to dynamos, port and starboard sides; 4½-inch steam to auxiliaries forward of the firerooms, starboard side; 3-inch steam to heating system, port side; 1-inch steam to galley and pantries, port side; 2-inch steam to whistle and siren, port side.

In each boiler room the distributing castings supply steam to the auxiliary feed pumps, fireroom, fire and bilge pumps, blower engines, ash hoists, tube cleaners and connections for blowing out sea chests.

Stop valves are located on the forward side of athwartship bulkheads. The distributing castings are anchored directly to the center-line bulkhead, and the expansion in the pipes and boilers is taken care of by expansion joints, proper allowance having been made for connections on bulkheads by making bolt holes oval.

The material of pipes and flanges is the same as for main steam lines. Where condensation is apt to accumulate, drains are fitted and piped to automatic traps which discharge to condensers and feed tanks.

The pipes are lagged the same as the main steam lines.

MAIN CONDENSER.

There is one main condenser in each engine room. They are 6 feet 7 inches in diameter by 15 feet 3 inches long by 13 feet between tube sheets. The shell plating is of steel, the water chests of composition; tubes of composition, 70 parts copper, 29 zinc, 1 tin; tube sheets of rolled Muntz metal, and the tube-supporting plates of sheet steel bushed with composition.

The forward water chest, being the one for the entrance and exit of the circulating water, has a division plate on the center line of the condenser. Sizes of inlet and outlet are $18\frac{1}{2}$ inches in diameter, the inlet nozzle being on the bottom.

DATA FOR ONE CONDENSER.

Number of tubes	4,878
B.W.G. of tubes	16
Outside diameter of tubes, inch	$0\frac{5}{8}$
Length of tubes between tube sheets, feet and inches	13-0
Number of supporting plates	2
Cooling surface, measured on outside of tubes, square feet	10,375
Ratio of cooling surface to total heating surface	.393

DATA FROM TRIAL.

	Starbd.	*Port*
Temperature of injection water, degrees Fahrenheit	44.0	44.4
discharge water, degrees Fahrenheit	101.0	101.6
Ratio of cooling surface to I.H.P. of main engines	1.05	

AUXILIARY CONDENSER.

There is one Blake horizontal combined air and circulating pump with surface condenser in each engine room.

DATA.

Diameter of steam cylinder, inches	6
air cylinder, inches	10
water cylinder, inches	10
Length of stroke, inches	12
Diameter of circulating-pump suction, inches	$5\frac{1}{4}$
discharge, inches	$5\frac{1}{4}$
air-pump suction, inches	4
discharge, inches	3
Cooling surface of condenser, square feet	550
Number of tubes	395
Outside diameter of tubes, inch	$0\frac{5}{8}$
Length of tubes, inches	104
B.W.G. of tubes	16

MAIN AIR PUMPS.

The main air pumps are of twin, vertical-beam type, of Blake design. They are located on the outboard side of the engine rooms, between the main condensers and the feed and

filter tank. The discharge is into the filter portion of the feed tank, and there is a $5\frac{1}{2}$-inch cross connection between the suction pipes from the main condensers.

DATA FOR ONE AIR PUMP.

Diameter of steam cylinders, each, inches	12
water cylinders, each, inches	30
Stroke, inches	18
Diameter of steam piston rods, inches	$2\frac{1}{2}$
air piston rods, inches	$2\frac{3}{4}$
suction nozzle, inches	11
discharge nozzle, inches	$9\frac{1}{2}$

DATA FROM OFFICIAL TRIAL.

	Starbd.	*Port.*
Average strokes per minute	19.0	18.0
vacuum, inches	28.0	27.9
I.H.P.	13.7	13.3

MAIN FEED PUMPS AND FEED SYSTEM.

There are two Blake vertical, plunger, double-acting, single, center-packed main-feed pumps in each engine room, located on the center-line bulkhead forward of the operating platform. The suction is from the feed-tank connecting pipe, and the discharge from the two pumps after uniting in one $5\frac{1}{2}$-inch pipe leads forward, passing through a grease extractor, which may be by-passed, and then through the feed-water heater, which also may be by-passed, into the after fireroom, reducing in size as each fireroom is passed.

DATA FOR ONE PUMP.

Diameter of steam cylinder, inches	13
water plunger, inches	$9\frac{1}{2}$
Stroke, inches	12
Diameter of piston and plunger rods, inches	$2\frac{1}{2}$
valve, inches	$4\frac{1}{2}$
suction nozzle, inches	$5\frac{1}{2}$
discharge nozzle, inches	$5\frac{1}{2}$
Material of water end	Composition.

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average double strokes per minute	26.5	31.5
I.H.P. per minute	59.0	61.0
pressure on feed main, pounds	325.0	325.0

FEED-WATER HEATER.

There is one horizontal feed-water heater on the discharge side of the main feed pumps, on the center-line bulkhead at the forward end of each engine room. The material of water heads is cast steel, and that of the tubes, tube sheets and shell, wrought steel.

DATA FOR ONE HEATER.

Total heating surface, measured on the outside of the tubes, square feet	830
Total number of tubes	756
Outside diameter of tubes, inch	00$\frac{5}{8}$
Length of tubes between tube sheets, feet and inches	6–09
B.W.G. of tubes	16
Thickness of tube sheets, inch	00$\frac{7}{8}$

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Temperature of feed water, degrees Fahrenheit	165.0	164.4
Pressure on feed line, pounds	325.0	325.0
Exhaust steam pressure in heaters, pounds	7.5	6.25

FEED AND FILTER TANKS.

The feed and filter tank is located on the outboard side of the after end of each engine room. It is rectangular in shape, and at the top is arranged as a filter, the filtering material being "loofa," which is placed between perforated plates. The filtering chamber is divided by vertical division plates, arranged so that the water flows under and over in succession. All material is steel, well galvanized before assembling.

Capacity of each feed and filter tank, gallons	5,475
Total capacity of feed and filter tanks, gallons	10,950

AUXILIARY FEED PUMPS.

In each starboard fireroom, located on the center line bulkhead, is one Blake, vertical-plunger, double-acting, single, center-packed auxiliary feed pump. Size of pumps, 11 inches by 8 inches, and 12-inch stroke.

The suctions are from the bottom blows, athwartship compartments, C. & R. pipes from ship's sides (after pump only), feed-suction pipe, reserve-feed water tanks, hose connection.

The discharges are to the auxiliary feed main, main and reserve-feed tanks (after pump only), hose connection, overboard, in port compartment, fire extinguisher.

FIREROOM FIRE AND BILGE PUMPS.

In each port fireroom, located on the center-line bulkhead, is a Blake, vertical-piston, double-acting, single, fireroom fire and bilge pump. Size of pumps, 11 inches by 8 inches by 12 inches.

The suctions are from the bilges, adjacent athwartship compartment, sea, C. & R. drainage manifolds, hose connection. The discharges are to the fire main, overboard in same compartment, hose connection, and fire extinguishers.

MAIN CIRCULATING PUMP AND ENGINE.

The main circulating pump and engine is located on the outboard side of the forward end of the engine room. The pump has three suctions, viz: the engine-room bilge, 18¼ inches; main drain, 15½ inches; and main injection, 18½ inches. The valves of the various systems are locked in such a manner that only one may be operated at a time.

DATA FOR ONE PUMP AND ENGINE.

Capacity of pump, gallons per minute		12,000
Diameter of suction nozzle, inches		18½
discharge nozzle, inches		18½
impeller, inches		56½
Width of tip of blade, inches		5¼
Diameter of H.P. cylinder, inches		10
L.P. cylinder, inches		20
Stroke, inches		10
Diameter of piston rods, inches		2¼
Length of connecting rods, center to center, inches		25
Valve travel, inches		3
Angular advance, H.P., degrees		33
L.P., degrees		34
	Top.	*Bottom.*
Steam lead, H.P., inch	$\frac{1}{16}$	$\frac{1}{8}$
L.P., inch	$\frac{5}{32}$	$\frac{3}{16}$
Cut-off, H.P., per cent. of stroke	0.77	0.715
L.P., per cent. of stroke	0.78	0.71

Diameter of H.P. piston valve, inches	$4\frac{1}{2}$
Size of L.P. slide valve, inches	$1\frac{1}{4}\times19\frac{1}{4}$
Diameter of crosshead pin, inches	$3\frac{1}{2}$
Length of crosshead pin, inches	5
Diameter of crank pin, inches	$4\frac{1}{2}$
Length of crank pin, inches	$6\frac{3}{4}$
Diameter of crank shaft, inches	$4\frac{1}{2}$
Length of bearings, inches	$5\frac{3}{4}\times8$ and 8
Crank angle, degrees	90
Area of crosshead slipper, square inches	60

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average revolutions per minute	140	133
I.H.P.	52.3	44.5

EVAPORATING AND DISTILLING PLANT.

The evaporator room is located amidships on the berth deck, forward of the engine-room hatches, the distillers being in the ventilator trunk to this room on a level with the main deck.

The plant consists of three evaporators and two distillers, having a designed capacity of 16,500 gallons of potable water in twenty-four hours. The distiller circulating pump is located in the forward end of the starboard engine room, fresh-water and evaporator-feed pumps in the evaporator room.

The evaporator shells and heads are of steel plate, the front steam heads for the coils being of cast steel and the back heads of composition. The tubes are brass, rolled into Muntz-metal tube sheets. Each evaporator is provided with steam gauges and relief valves for the shell and coils, also water column, gauge cocks and blow-off valves.

The distillers are vertical and of the Bureau type. The shell is of cast iron, the covers and tube sheets of composition, and tubes of brass, tinned inside and out. The tubes are expanded into tube sheets by rolling, and expansion is taken care of by having the bottom tube sheet made with a flange that works in a stuffing box. The circulating water enters at the bottom, passes through the tubes and is discharged at the top; the vapor enters the shell at a point close to the top

tube sheet, the water leaving at a similar point near the bottom tube sheet. Baffle plates are provided for the inlet nozzles for the circulating water and vapor.

The following pumps of the Blake type are provided: Distiller circulating, size, 12 by 14 by 12 inches; distiller fresh-water, size, 4½ by 5 by 6 inches; evaporator feed, size, 4½ by 5 by 6 inches.

Piping.—Steam for the evaporators and pumps is taken from a reducing valve located in the evaporator room at the terminus of a branch from the auxiliary steam line. The vapor may be sent to the distillers or the auxiliary exhaust line, the evaporators being arranged for operation in single effect. The fresh water from the distillers passes through a meter into a small reservoir tank located in the evaporator room, from which it may drain by gravity to the main and reserve-feed tanks or may be pumped into these tanks or the fresh-water tanks. The distiller circulating pump has an independent connection to the sea, and, in addition to discharging to the distillers, may discharge to the sanitary system or fire main. The evaporator-feed supply is taken from the suction side of the distiller circulating pump.

A test was made of the plant when the ship was lying at anchor in the harbor at Delaware Capes, December 6, 1906. First each evaporator and distiller was given a two-hour trial, and then the entire plant was run for two hours, thereby furnishing an equivalent full-capacity test of four hours. The discharge from the distillers was run into the reservoir tank and a careful record kept of the number of full tanks. The water was tested for purity at frequent intervals with a solution of nitrate of silver, the concentration of water in the evaporators every half hour, and all other conditions recorded every fifteen minutes.

The following are the averages of the data obtained from the two-hour test of entire plant:

Steam pressure in evaporator coils, pounds	58
shells, pounds	12.5
at reducing valve, pounds	122
Pump strokes per minute for entire plant:	
Evaporator feed pump	24
Distiller circulating pump	32
Output of distilled water, gallons	1,619
Capacity of distilled water in twenty-four hours, gallons	19,428

DATA FOR ONE EVAPORATOR.

Heating surface, tubes only, square feet	277
Gallons of water evaporated per square foot of heating surface in twenty-four hours	20.97

DATA FOR ONE DISTILLER.

Cooling surface, tubes only, square feet	165
Gallons of water distilled per square foot of cooling surface in twenty-four hours	58.84
Gallons of water distilled in twenty-four hours	9,714
Temperature of sea water, degrees Fahrenheit	42
fresh-water discharge from distillers, degrees Fahrenheit	94
external air, degrees Fahrenheit	44
evaporator room, degrees Fahrenheit	100
Concentration of water in evaporators by hydrometer test, degrees Fahrenheit	$0\frac{2}{32}$
Quality of water distilled	Good.
Height of water in evaporators above bottom of glass, inch	1

REFRIGERATING PLANT.

The refrigerating plant is located on the starboard side of the berth deck forward of the forward turret, the refrigerating rooms, six in number, being aft of it, and the scuttle butts on the gun deck. The rooms are isolated by air locks and are insulated with cork covered on the inside with zinc, except the floors, which are covered with sheet lead.

There are two ice machines of the Allen dense-air type, each designed for a cooling effect equivalent to two tons of ice in twenty-four hours. The piping for the cooling air is arranged so that it may be sent through the refrigerating rooms and returned to the machine, or through the scuttle-butts, returning to the machine, or through the ice-making tanks before going to the refrigerating rooms.

During the test of the plant the air was sent through the ice-making tanks, then to the refrigerating rooms and scuttle butts in succession.

DATA FROM TRIAL.

Duration of trial, hours	8
Temperature of water to be frozen, degrees Fahrenheit	53
circulating water, degrees Fahrenheit	39
atmosphere, degrees Fahrenheit	41
room, degrees Fahrenheit	73
water in scuttle butts, degrees Fahrenheit	62
drawn from scuttle butts (at end of three hours), degrees Fahrenheit	54
Quantity of ice in twenty-four hours, pounds	950
water cooled in twenty-four hours, gallons	2,376
Initial temperature of refrigerating rooms, degrees Fahrenheit	60
Final temperature of admiral's room, degrees Fahrenheit	21
captain's room, degrees Fahrenheit	22
ward-officers' room, degrees Fahrenheit	8
junior officers' room, degrees Fahrenheit	17
warrant officers' room, degrees Fahrenheit	22
crew's rooms, degrees Fahrenheit	23
Total wall area of rooms, square feet	1,161
cooling surface of coils, square feet	521
Steam pressure, pounds	52
Compressor-cylinder pressure, pounds	230
Expander-cylinder pressure, pounds	52½
Revolutions per minute	118

MACHINE DATA.

Diameter of steam cylinder, inches	9¾
compressor cylinder, inches	7
expander cylinder, inches	5¾
piston rods, inches	1⅛
Length of connecting rod, center to center, inches	26⅝
Stroke, inches	13

Electric Plant.—The electric plant consists of eight 100-kilowatt generating sets (125-volt pressure at the terminals), direct connected to engines of the vertical, cross-compound, double-acting, enclosed type. Generators and engines are of the General Electric Company manufacture.

Four of these sets are located in each of the two independent dynamo rooms, one forward and one aft, on the upper-platform level.

In each dynamo room there is a main dynamo-switch board which controls the generators in their respective rooms.

Steam is taken from the auxiliary steam line, and each dynamo plant has a combined air and circulating pump and condenser for the exclusive use of the dynamo—that for the forward plant being located in a pocket forward of the boiler compartments, and that for the after plant in the passage on the starboard side between the engine and firerooms. There is a steam separator in each room, which drains to a trap and the steam end of the air pump. The exhaust is into either the dynamo condenser, main or auxiliary condenser or the atmosphere.

Adjacent to each dynamo room there is a distribution room in which is installed a distribution-switch board which is energized from either of the two main dynamo-switch boards. From the distribution-switch boards all the feeders which supply energy to the lighting and power systems are led.

The lighting system comprises a total allotment of 1,100 electric fixtures, not including special outlets for signal lanterns, truck lights, Ardois signal sets and outlets for special fixtures. There are two arc lamps in each of the engine rooms and firerooms.

There are six 30-inch searchlights, two being installed on the forward bridge, two on the after bridge and one on the searchlight platform on each of the masts. Separate and distinct from the lighting system is the power system, which supplies energy to all auxiliary machinery run by electric motors.

Included in the power system are the following: Fourteen 3-inch ammunition-hoist motors; twelve 7-inch ammunition-hoist motors; four ammunition conveyers; thirty-three ventilator fans; four pumps; six deck winches; two boat cranes; laundry machinery; workshop machinery; two 12-inch turret-turning equipments; four 8-inch turret-turning equipments; two 12-inch turret-power equipments; four 8-inch turret-power equipments; forty-one power doors, and five power hatches.

A complete system of wireless telegraphy is installed. Besides the lighting and power system there is an interior-communication system, which includes call bells, general alarms, shrill whistles, fire alarms, voice tubes, telegraphs and indicators.

A system of mechanical telegraph is also installed for communication between the bridge and engine room.

DATA.

Type of engines	Vertical cross-compound.
Number of revolutions per minute	350
Steam pressure, pounds	150
Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	18
Length of stroke, inches	10
Diameter of piston rods, inches	2¼
Number of generators	8
Capacity of each generator, in kilowatts	100
Dynamos	10 pole, 125 volts.

DATA FOR ONE DYNAMO CONDENSER.

Make	Blake.
Type, Horizontal, combined air and circulating pump, with surface condenser.	
Diameter of steam cylinder, inches	10
air cylinder, inches	14
water cylinder, inches	16
Length of stroke, inches	12
Cooling surface of condenser, square feet	970
Number of tubes	800
Outside diameter of tubes, inch	00⅝
Length of tubes, inches	91
B.W.G. of tubes	16

ASH HOISTS.

The ash-hoist engines were designed and built by the contractors. They are located entirely within the ventilator trunks, and are designed to stow in one corner of the trunk within a plane not exceeding twelve by eighteen inches. The hoist will deliver to and may be operated from either the main or upper decks, the bucket-guide rails extending from the bottom of the ventilator trunk to a point well above the upper deck, with trolleys at the main deck for delivering ashes to

the chutes at the ship's side. On the test 300 pounds were hoisted from the fireroom floor to the upper deck in five seconds, including starting and stopping, with 100 pounds steam pressure.

The principle of the engine is a long cylinder taking steam at both ends through a piston valve that is controlled by the operator through the medium of a spiral spindle worked by a wheel and gearing outside the ventilator trunk. The spiral spindle works in a nut carried by the crosshead, thus putting the valve automatically on the center and stopping the engine when the operator stops turning. At the proper height above the steam cylinder is the top sheave bracket, between which, on guide rods, the crosshead travels. This crosshead is fastened to the piston rod and carries multiple sheaves corresponding to those carried by the top bracket, that give a maximum bucket travel of 50 feet for a piston travel of 6 feet 9 inches.

DATA FOR ONE ENGINE.

Diameter of cylinder, inches	7
Stroke, inches	81
Diameter of piston valve, inches	$1\frac{1}{2}$
Travel of piston valve, inches	2
Diameter of piston rods, inches	$1\frac{5}{8}$
rope sheaves, inches	8
Pitch of spiral spindle, inches	13
Number of rope sheaves	3
ash-hoist engines in ship	6

FORCED-DRAFT BLOWERS.

The forced-draft blowers are located in the firerooms, being suspended from the protective deck about over the center of the stokehold. The engines and fans were designed and built by W. D. Forbes & Co.

The arrangement is the same for all of the firerooms, there being two engines, each driving a fan located athwartship, so that the air is discharged directly to the stokehold floor at a distance of about two feet from the front of the boilers. The supply is taken from the fireroom ventilator ducts, which are closed at the bottom when under forced draft.

DATA FOR ONE BLOWER ENGINE.

Number of cylinders	2
Diameter of H.P. cylinder, inches	5
L.P. cylinder, inches	10
Stroke, inches	6
Diameter of H.P. valve, inches	2
L.P. valve, inches	4
Valve travel, inches	$1\frac{1}{8}$
Diameter of piston rods, inches	$1\frac{5}{8}$
valve stems, inch	$0\frac{3}{4}$
crank shaft, inches	3
pins, inches	3
fan shaft, inches	$2\frac{3}{4}$
Length of connecting rod between centers, inches	15
Number of bearings	4
Total length of bearings, inches	$22\frac{1}{2}$
Crank angle, degrees	180
Base area of engines assembled, inches	30×30
Height of engines assembled	$54\frac{1}{2}$

DATA FOR ONE BLOWER FAN.

Diameter of fan, inches	66
Number of blades	6
Width of tip of blades, inches	15
Area of induction nozzle, square inches	1,017
eduction nozzle, square inches	1,385

DATA FROM OFFICIAL TRIAL.

Average revolutions per minute	314
air pressure in firerooms	.596
I.H.P. for one blower	8.13
twelve blowers	97.6

HEATING SYSTEM.

The heating system is divided into two sections, one forward and one aft, the forward section being divided into five radiator circuits and one galley and pantry circuit, the after section being divided into six radiator circuits and one galley and pantry circuit. Steam for radiators is supplied from the auxiliary steam line, the connection at the line being fitted with pressure gauges, stop and relief valves and reducing valves, set at 30 pounds. From this point the steam is carried to distributing manifolds and divided into circuits, as

mentioned above, each circuit being arranged with a stop valve so that it can be operated independently of the others.

The galley and the pantry circuits also receive steam from the auxiliary steam line, and are fitted with valves and gauges as on radiator lines, but are entirely independent, being constantly under steam.

The circuits supply steam to laundry, bakery, galleys, pantries, water heaters in bath and wash rooms, and also a coil located in fresh-water tank to prevent water from freezing.

The drains from radiators are also divided into circuits and lead to four receiving manifolds, two being located forward and two aft, each circuit being arranged with a check and stop valve to prevent water from one circuit backing up into another. The discharges from manifolds lead to traps in engine rooms, each manifold having an independent trap, and the drains from these traps are connected together and lead to a drain manifold. This manifold has two outlets, one leading directly to filter tank and the other to a manifold which discharges to main condenser, auxiliary condenser or filter tank at will.

The drains from galleys and pantries lead directly to traps in engine rooms, each section being provided with an independent trap, and these traps discharge to drain manifold mentioned above. The drains from water heaters in bath rooms have no connection whatever with other drains, and are carried directly to the auxiliary-exhaust lines.

The circuit steam and drain pipes consist of seamless-drawn brass pipe, iron-pipe size, all pipes up to 1½ inches being connected together by composition fittings, but above that size composition flanges are used. Connection at watertight bulkheads are made with composition stuffing boxes, and copper U bends are installed throughout the lines to provide for expansion.

Particular care has been taken to prevent water hammer, and as all drains lead downward no pockets have been formed.

The radiators consist of coils made up of 1-inch seamless-drawn brass pipe, iron-pipe size, and composition fittings, stop and air valves.

The stop valves have stems of triangular cross section, valve-stem guards, socket keys for opening and closing, and unions on outlet end of valves so that radiators can be taken down without disturbing the lines.

Large radiators are divided into sections, each section having an independent steam and drain valve, so that in mild weather it is not necessary to have steam on the entire radiator.

MACHINE SHOP.

The machine shop is located amidships on the berth deck, forward of the after 12-inch barbette and between the machinery hatches, access to it being from the athwartship passage, and from the engine rooms through their respective entrance hatches. The machines are arranged with line and counter shafting driven by a 15-horsepower General Electric motor. On a test made to determine whether the motor was sufficiently powerful to do the work, with all machines working on hard cast iron at a heavy cut, 6.21 horsepower was developed.

The following machines are installed:

1 28-inch extension gap lathe;
1 14-inch engine lathe;
1 16-inch crank shaper;
1 31-inch radial drill press;
1 16-inch sensitive drill press;
1 30-inch grindstone;
1 universal milling machine;
1 12-inch emery grinder;
1 combined hand punch and shears;
6 bench vises.

The tools are provided with the most modern attachments, including scroll and drill chucks, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

NOTES.

THE SIZE OF WARSHIPS.

Naval debates in the House of Lords are most academical in character, but they are often not less informing on that account; indeed, compared with the political bickerings that are often elsewhere intruded into naval discussions, the calm examination of facts that takes place is a distinct gain. So long as the First Lord of the Admiralty is a Peer, and the voting of money rests with the Commons, the naval interest is, to a certain extent, divided, and we may look for debates in the Upper House similar in character to that which took place on Wednesday of last week. Lord Brassey, in raising the discussion, "had no fault to find with the administration." "The Opposition," he tells us, "is silent because they are satisfied with the general administration of the Navy;" and, "the supporters of the Government know that the strength of the Navy in men and ships is being adequately maintained." Moreover, "where economy has been possible votes have been cut down, and no pains have been spared to perfect the training of the Navy." What more a grateful country could ask from its rulers it is difficult to imagine; still, there are one or two things that might be inquired into with advantage; and Lord Brassey forthwith proceeded to inquire into them.

First there was the case of the *Dreadnought*. Her design had been "commended by an overwhelming weight of authority" at home, and had received still more practical recognition abroad, being followed by nearly every maritime Power—by Russia, Japan, France and the United States. Still, the *Dreadnought* cost something like two millions sterling, and such ships must be few. There were operations in war which

could be more effectively performed by many ships than by few. Admiral Sir Reginald Custance has said, "No ship, however large, can stand up against the fire of two or three battleships." It is a sweeping assertion, as it would be to say, "No small ships, however numerous, can stand up against the fire of one large battleship." Before we can reduce such maxims to their current value we must define the limits of "small" and "large." The problem, which will never be settled in peace time, at any rate, is whether two one-million-pound battleships can be made equal to one two-million-pound battleship; or whatever the suggested proportion may be.

The balance of opinion is, however, evidently against subdivision of power, for battleships have been getting bigger and bigger since the days of the *Great Harry*, and more and more costly. Concentration is still a cardinal principle in sea fighting, as on land. A *Dreadnought* engaged with two one-million-pound ships has but to disable one for victory to be in her hands. It is true that in the operation she may receive damage herself, but the big ship will have thicker armor and higher speed; and, moreover, there is no chance of her power being divided, as may be the case with the two ships, which may lose the advantage of concentration. Sir William White, who has not in practice been averse to big battleships, if one may judge by his creations, has suggested that the armament of the *Dreadnoughts* might be distributed between three vessels equal in speed, armament and defence, and possibly superior in fighting power to the larger ships. The question arises; Can the three hulls and machinery, equal in efficiency to two *Dreadnoughts*, be built for the same expenditure? The guns and their mountings involve a large part of the cost of the *Dreadnought*—about 30 per cent. of the total; and it is well known that the cost of the hull of a ship does not decrease *pro rata* with size, nor does the power and cost of propelling machinery decrease with the displacement weight, the speed being constant.

No one, we think, would be more ready than the great

naval constructor to acknowledge that considerable sacrifice must be made by replacing two units by three, apart altogether from the financial question. Sir William apparently holds that the sacrifice would be warranted; and it might, of course, take one of many directions: the smaller ships would need greater power for equal speed, and the weight of hull structure would be greater for the same armament, whilst thicker armor would be more easily provided for the bigger ships. If we take cost as a basis of comparison, the advantage in these respects lies with the two vessels as against the three; and in many other things, such as up-keep, coal endurance, etc., the big ships are to be preferred. In regard to maneuvering in battle, there is more room for question, and it is a question the naval tactician must decide.

On the other hand, two fortunate shots from mines or torpedoes would annihilate the whole force if it consisted of two larger ships, whilst in like case of three smaller ones, there would be a ship left; but against this, three ships are more likely than two are to hit a floating mine. Much the same thing may be said of strandings or other accidents of navigation. In strategy the question of size is not easy to decide. Two ships can only be in two places at once, whilst it might be necessary to occupy three, or it might be desirable to proceed to a rendezvous by three routes. As battleships are, however, primarily intended for the line-of-battle, this is not so important a consideration as with cruisers. The big ship will, however, have higher speed and wider radius of action than any number of smaller ones, other features of design being proportionate.

For cruisers the problem of design is even more complicated than it is for battleships, as the duties of the former are more varied. We have gone on from the days of the *Blake* increasing the size and power of cruisers, until many of them have become really modified battleships; but naturally the cost has gone up also. A cruiser that costs about a million and a half is evidently too expensive a vessel to carry out the duties of the old frigates. Lord Tweedmouth, repeating Lord

Brassey, says the protected cruiser is almost a non-combatant; but the seme might be said of all vessels in regard to others appreciably stronger than themselves. The old sailing frigates, which were never classed as non-combatants, were not supposed to stand up to battleships, although on one occasion a frigate did fire on a battleship, and was promptly sunk for her impertinence. Lord Brassey is satisfied that the British Navy has an overwhelming superiority in protected cruisers; as, indeed, it should have, considering the overwhelming superiority of the maritime commerce it would have to protect. Lord Tweedmouth, who naturally supports the big battleship, says, in quoting naval opinion, that the big armored cruiser is more likely to be of advantage in war than the smaller cruiser; and he is of opinion that no foreign cruiser would be able to stand up against the *Invincible*. One may readily agree with this, but it does not follow, as appears to be thought by some, that we should not have cruisers capable of standing up against weaker vessels than *Invincibles*. Naturally, we would like all our cruisers to be *Invincibles*, if we could afford it. In some respects the cruiser problem is the reverse of the battleship problem, inasmuch as dispersion is a virtue with this class of vessel.

Lord Brassey has no good word to say for the eight scouts that have lately been built. They are, he says, costly but not satisfactory; flimsy in construction, and weak in armor and armament, speed being their only merit. He also disapproves of the special torpedo-boat destroyer *Swift*, which is being built by Messrs. Cammell, Laird & Co., the estimated cost of which is over a quarter of a million. This vessel is to be 345 feet long by 34 feet 2 inches beam, and is estimated to have a displacement of 1,800 tons. With 30,000 horsepower she is designed to steam at 36 knots, and will carry 180 tons of oil fuel. Lord Brassey evidently considers that the usefulness of such a vessel will not be equal to her cost, and would prefer two ocean destroyers, which could be built for about the same money. It certainly cannot be claimed for craft costing a quarter of a million sterling that numbers can be cheaply

expended in time of war. The advance in torpedo craft is, however, only characteristic of that in all vessels of the fleet, from pinnaces to battleships. In default of a design for an effective cruiser for scouting duties, which Lord Brassey says "the ablest constructors, both in the Navy and outside," have failed to produce, he would call attention to the subvention of mail steamers. Lord Brassey has always taken an interest in naval auxiliaries ever since he raised the first naval volunteer corps, about thirty-five years ago, and the principle of merchant-steamer subvention owes much to his efforts. He quoted Admiral Lord Charles Beresford's evidence before the Committee on Subsidies in support of his views.

Auxiliaries are, however, not generally considered with favor by the Sea Lords. The Royal Naval Artillery Volunteers, an enthusiastic and hard-working body, were disbanded with scant courtesy some years ago; and the present body cannot be said to have received enthusiastic support. Lord Tweedmouth now says that mercantile cruisers "can be relied on very doubtfully," as "they must be extremely fast, or they will not be able to escape from ships of more formidable type." It would be instructive to compare the maintained sea speed and coal endurance of the protected cruisers—which are, the First Lord says, "almost non-combatant ships"—with the class of ocean liners that would be available for scouting duty and conveying information in time of war. No doubt wireless telegraphy has completely modified the practice of scouting in the present day, but speed and coal endurance are still essentials to effective service of this nature. We have yet to learn how far "wireless" will stand the rough test of war. The First Lord, however, acknowledges that "if any mercantile cruisers are to be capable of being of service in war," they would be vessels like the two new Cunarders.

The most important feature of the debate was, however, the explicit statement by the First Lord of the Admiralty as to the policy of his Government. Fears have been expressed—they may have been founded on rumors originally circulated

for political purposes—that the present Administration was careless of the position the country has held as a naval power. The rumors at first gained some color from the reductions made in the Navy Estimates. It must be remembered, however, that these reductions did not originate with the present Administration, and there is nothing whatever to show that they are due to political subserviency on the part of the Board of Admiralty. Although a clear political horizon may never be taken as a reason for slackness in preparation for national defence, it is but reasonable to put forth additional effort if complications abroad threaten. A few years ago—the period of our "splendid isolation"—such complications did threaten, and the Navy Estimates went up by leaps and bounds. Since then the alliance against us, which was mostly to be feared, has ceased to threaten. The purchase of two powerful warships on the eve of the Russo-Japanese war swelled the expenditure of the year by a large amount, and might be taken as a reasonable cause for some reduction to follow.

Making all allowances, however, a reduction of the Fleet below the strength needed to maintain our naval supremacy would be a dangerous experiment that the country would justly look on with alarm; but this Lord Tweedmouth in the Lords, and Mr. Robertson on the same day in the Commons, assure us is not to take place. The First Lord of the Admiralty said that the axiom must be accepted by everyone "that our country must at all events keep the command of the sea, without having any regard to the cost at which that may be done." Mr. Robertson, speaking somewhat later on the same day in the House of Commons, said: "As to the two-Power standard, I would say—and I believe it has already been stated elsewhere by the First Lord of the Admiralty—that it is the determination of the Government to maintain our naval supremacy as it stands now, and in the event of the Hague Conference proving abortive, to hold ourselves ready to take steps to maintain our relative position."—"Engineering," April 26, 1907.

LIQUID FUEL FOR STEAMERS.

The steamer *Elax*, one of the well-known fleet of oil-carrying vessels owned by the Shell Transport and Trading Co., Ltd., which has undergone extensive repairs and overhaul by Messrs. Smith's Dock Co., Ltd., has had an official trial and demonstration of the efficiency of the liquid-fuel-burning installation fitted by the latter firm. The trial proved eminently satisfactory to all concerned, steam being easily maintained at the required pressure to propel the vessel at a speed of 11 knots on a consumption of 14½ cwts. of liquid fuel per hour. This is equivalent to a consumption of only 17½ tons per day, for all purposes, as against 30 tons of coal per day consumed for the same work on her previous voyages, a fact of considerable interest to shipowners and engineers. The *Elax*, one of the medium-sized vessels of the Company, is 4,015 tons gross register, and capable of carrying some 5,500 tons of cargo, and is the eighteenth steamer of this kind fitted for burning liquid fuel. The installation fitted by Smith's Dock Company includes the Rusden-Eeles patent burners, and the Flannery-Boyd patent system of heating coils and gravitation tanks for purifying the fuel before being conducted to the burners.—"Page's Weekly."

A LESSON IN MARINE-ENGINE DETAIL.

One sometimes wonders, when one hears of serious results springing from small causes, if we shall ever get through the elemental stage of marine-engine practice. Sea-going engineers would do much towards reaching this desired end if they would oftener take steps to put on public record the mishaps which occur to machinery under their charge, and the way in which accidents are remedied. An excellent example of this nature is given in the current number of that useful service publication the "Royal Naval Artificer Engineers' and Engine-Room Artificers' Review." The detail that gave trouble in two of His Majesty's ships was the keep-ring; the particulars given are as follow: The *Argonaut*, first-class

cruiser, homeward bound from China, reached Malta last September, and on leaving that port proceeded to carry out a passage trial. The trial commenced with 125 revolutions, 18 knots being logged. After the run had continued for some time there was a loud report in the starboard engine room. The engineer artificer at once gave the order "Center the links," and the next moment the engine room was full of steam; whilst the senior engineer rushed on deck and closed the master-valve to that set of engines. All men were cleared out and the watertight doors were closed. The port engines continued revolving at 130 revolutions. It was found, ultimately, that the mishap had occurred in the forward low-pressure cylinder. Nothing could be seen externally, but on moving the engines very slowly a clicking noise was heard with each stroke of the engine. The steam which filled the engine room was accounted for by the high-pressure receiver valve lifting when the links were centered. It was thought at first that the spring-ring was fractured, but a card taken was normal. The starboard engine was kept going until nearing Gibraltar, when the cylinder was opened up, and it was found that half the keep-ring had become displaced, having been carried into the port, where it was sheared off, thus accounting for the report.

The second mishap of this nature reported in the same publication occurred a short time previously to a smaller vessel. The third-class cruiser *Flora* was cruising at 48 revolutions when the starboard high-pressure engine gave a succession of heavy knocks. The engines were eased, and the trouble had apparently disappeared, but on increasing to above 70 revolutions, it was renewed. As the ship was in company with the *Cambrian*, another third-class cruiser, the port engines were increased to 95, the starboard engines not being allowed to exceed 70, revolutions. On opening up, it was found that the high-pressure piston keep-ring had quite disappeared. Further search revealed parts of it, with one stud, in the steam port, other parts were jammed on the exhaust side of the slide-valve and the strengthening vanes of the

valve, whilst some pieces were found in the eduction pipe and the exhaust side of the medium-pressure slide valve. There were in all 33 pieces, varying from 1 inch to 10 inches long. The ring was $1\frac{3}{8}$ inches by $\frac{3}{8}$ inch. The actual damage done was a slight roughening of the exhaust edge of the high-pressure slide valve, whilst the medium-pressure slide valve had a large piece of the baffle broken off. This was found in a pocket just below the mouth of the medium-pressure eduction pipe.

A keep-ring is not, of course, a part actually necessary to the working of an engine, it being fitted as a matter of precaution to prevent accident. When, however it is not securely fastened its acts in a distinctly opposite manner, as these two accidents show. One cannot but admire the courage of the chief engineers of both these ships in not advising their respective captains to have the engines stopped—they are both twin-screw vessels—for an immediate examination. No one, in either case, knew what was the matter, though something of a serious nature had undoubtedly occurred; and this might, at any time, have resulted in no inconsiderable disaster. We must, however, confess that had such an incident occurred to a vessel that was our property, we should have preferred a more cautious procedure, and our leaning that way would have been much strengthened had we been on duty in the engine room. Even if it had been known that it was only the keep-ring, it would not have been much more reassuring, for certainly one could hardly expect that substantial pieces of metal would go through two engines in the manner described without doing very much more damage. However, officers and men of His Majesty's ships are expected to run more risks than mere civilians; and we know it is a point of honor in the engine room—and a very salutary one—not to ask to stop the engine excepting under dire necessity. It would seem that there is need for a more stringent general order that all mishaps of a serious or threatening nature should be inquired into immediately, unless there should be present some extremely urgent reason to the contrary.

In the case of the *Argonaut* a new ring was made, and the voyage was continued across the Bay of Biscay. In the original ring square studs had been fitted in the piston, corresponding holes being made in the keep-ring, the latter being secured by $\frac{3}{16}$-inch split pins. One would have thought, considering the light, though important, duty the ring is called upon to perform, this would have been security enough; and this was evidently the opinion of the Admiralty and the contractors—the Fairfield Company. Events, however, showed this not to have been the case, and probably the split pins were sheared through by constant repetition of stress due to the inertia of the ring. In the case of the *Flora*, the engines for which vessel were made at Barrow, the ring was secured to the junk ring by five $\frac{5}{8}$-inch square-headed studs, and the former was held in position by split pins. When a new ring was made the edge was reduced to $\frac{1}{4}$ inch in thickness, and this gave space to allow nuts to be screwed on the studs and secured by split pins. What precaution was taken to secure the new ring of the first-class cruiser's engine is not apparent from the narrative; but it is advised that the studs should be left longer, and should be screwed above the square to receive a nut, as well as a split pin.—"Engineering," Feb. 22, 1907.

THE MARINE STEAM TURBINE.

The various successful trials of turbine steamships made during the past twelve months have generally been hailed as further triumphs for the marine steam turbine. In many respects this conclusion is entirely justified; but, at the same time, it must be confessed that the trials left matters pretty much where they were before. The pioneer turbine boat, the *Turbinia*, established once for all that the turbine could meet and beat the reciprocating engine when applied to the propulsion of high-speed craft. Such vessels must, however, always form but a small, if important, minority of the total tonnage of the world, and we seem no nearer than we were eight years ago to the solution of the problem of adapting the turbine to

the propulsion of even the larger size of tramp steamship From the outset it has been recognized that the difficulty has lain with the propeller. Even in the case of high-speed boats the turbine has had to be run much below its most economical speed, and this has been the case although its dimensions have been enormously increased as compared with those of equal output applied to the driving of electric generators. The maximum peripheral speed of the buckets in marine work is commonly much less than half that of a turbine employed for electric lighting; and if the same general design were followed in the two cases, the turbine should be at least four times as long if equal economy were to be maintained. Moreover, in order to get this relatively low peripheral speed at a moderate number of revolutions per minute, the diameter of the turbine has to be increased, which again involves a marked augmentation in the weight of the machinery per horsepower developed. The designers have gone in this direction nearly as far as they dared, and as a consequence the total weight of the turbine machinery fitted to steamships seldom shows an advantage of more than 5 per cent. as compared with reciprocating engines. In electric-light practice the turbine unit often weighs only one-fifth to one-sixth as much as the corresponding reciprocating unit. In spite of the relatively high weight of the marine turbine, it is nevertheless, as stated, run in general much below its most economical speed; and here undoubtedly is to be found one reason for the disappointing results which have in some cases been realized in service. In no instance, we believe, has the turbine engine failed to beat, both in speed and economy, a sister reciprocating boat on the measured mile. In service, however, the latter has, in more than one instance, proved the less expensive in the matter of fuel.

As the difficulty with the turbine is to get its speed down, designers have naturally selected as their speed for calculation that aimed at on the trial trip. With turbines in which the speed is maintained constant at all loads, as in those employed for electric lighting, the hydraulic losses below the governor-valve remain a nearly constant proportion of the useful work

done, so that throttling the steam to diminish the output some 20 per cent. or so means no serious reduction in the economy. If, however, at the same time that the steam is throttled the speed of the turbine is reduced, additional losses, arising from hydraulic shock, are added to those due to throttling, and the steam used per indicated horsepower per hour may very rapidly augment, particularly if, as is invariably the case in marine practice, the turbine is much underspeeded, even at full power. It would appear, therefore, that specifications for turbine boats should lay much more stress on the economy to be attained during service conditions than on the full-speed trial, and the combination of turbine and propeller should be such as to give the best economy at the relatively low-service speed, even if this involves the sacrifice on the trial trip of the fraction of a knot or so.

An additional reason for the less favorable showing of turbine craft at sea was advanced by Professor Biles in his lectures to the Royal Scottish Society of Arts last winter. He notes that at sea the resistances corresponding to a given speed appear to be greater than in trial-trip conditions. This is not a serious matter in the case of craft fitted with reciprocating machinery, since in general with them the propeller is of ample proportions for the thrust it has to develop. With turbine craft, however, the propeller is often of such scanty proportions that quite a moderate increase in the thrust to be developed leads to cavitation and a consequent large diminution in its efficiency. The recent careful experiments made at the Navy Tank, Washington, D. C., have much simplified the problem of combining a turbine and propeller so as to obtain the maximum possible over-all efficiency. Many people are still under the impression that the percentage of slip is a direct measure of the propeller losses. In the old days this view led to the production of fancy propellers in great variety, all schemed out with the idea of reducing the slip to a vanishing quantity. But these propellers invariably proved less efficient than those they were designed to replace, and it was then pointed out by Rankine that some slip was necessary if any thrust at all were

to be developed. In spite of this, however, the false impression as to the significance of slip still survives, and promises to continue popular for many years to come.

The Washington experiments demonstrate once again that a high slip ratio, even as much as 40 per cent., is no bar to a reasonable propeller efficiency, which, even with the figure stated, may be 66 to 67 per cent. It is not improbable, indeed, that this figure might be improved on by giving the propellers an increasing pitch from the front edge to the back. It is known that at low slip ratios there is no appreciable advantage in this procedure, but theory indicates that there might be a substantial gain when the slip approaches 30 per cent. or more. One drawback to the practice would be that the efficiency in going astern would be diminished, and turbine craft are already somewhat deficient in the matter of retrogression.

Although, as stated, the Washington experiments have simplified the problem of selecting the propeller most appropriate to a given set of conditions, they have pretty conclusively demonstrated the hopelessness of any such improvement in the propeller as will permit of the ordinary reaction steam-turbine being applied to the driving direct of the propellers of the average tramp steamer. If the turbine is to come into general use for such craft, either some form of reduction gearing must be used, or some other form of turbine adopted. An analagous case is found in those water-power plants in which the quantity of water available is small, whilst the head is high. In such cases reaction turbines are never used, as the requisite speed of rotation becomes enormous, but some form or other of impulse turbine working with partial admission. With water such turbines have very high efficiencies; but partial admission in the case of a steam turbine involves some serious losses, since the wheel has to run downward. Nevertheless, it is perhaps on these lines that the application of the turbine to small-powered craft will ultimately be developed.

At first sight the system of partial-admission impulse tur-

bines, velocity compounded on the Ferranti or Curtis system, would appear to possess great advantages for marine work in all cases. It is, however, impossible to use as narrow blades with velocity-compounded turbines as with the pressure-compounded type. With the former, provision has to be made for a very great increase in the radial depth of the belt of steam as it flows through the blades, and if the latter are very narrow, this radial expansion, which, it must be remembered, occurs without change of pressure, would be effected so abruptly as to lead to very large eddy losses. Here, no doubt, is to be found the reason for the failure, so far, of this type to show any saving over the Parsons marine pattern in the matter of the weight and length requisite to the attainment of a reasonable economy of steam. In our issue of November 16 we reproduced a paper by Dr. Lasche, showing a marine turbine of this type designed to develop 8,000 horsepower at 400 revolutions per minute. The diameter over the tips of the low-pressure blades was 2¼ meters (7.37 feet), and the length of the working part, from admission to exhaust, about 8 feet 6 inches. The estimated steam consumption (the turbine was never built) was, we understand, about 14 pounds of steam per brake horsepower hour. This is no doubt an excellent result in the matter of compactness, combined with a reasonable efficiency; but there would be little difficulty in equalling it with an ordinary reaction turbine having properly-set blades, and the weight of the latter would be substantially less. No doubt the leakage loss at the high-pressure end of the reaction turbine would be considerable, but would not exceed that due to the extra hydraulic resistances which are inseparable from the system of velocity compounding adopted for the corresponding high-pressure end of the Allgemeine turbine. For larger powers, therefore, we think the Parsons turbine will hold its own; but there does seem a possibility that a combination of the Curtis arrangement for the high-pressure end with the Parsons for the low-pressure might possess advantages in the case of the lower powered and relatively slower vessels. A very detailed study of the

problem would, however, be necessary to definitely settle this, as in steam-turbine practice the aphorism that "general reasoning is generally wrong" acquires an exceptional validity. —"Engineering."

LIMITS OF THERMAL EFFICIENCY IN INTERNAL-COMBUSTION ENGINES.

At the ordinary meeting of The Institution of Civil Engineers, on Tuesday, the 26th February, Sir Alexander Kennedy, LL. D., F. R. S., President, in the chair, the paper read was "On the Limits of Thermal Efficiency in Internal-Combustion Engines," by Dugald Clerk, M. Inst. C. E. The following is an abstract of the paper.

The Institution Committee on the Standards of Efficiency of Internal-Combustion Engines among their recommendations as to the standard engine of comparison for internal-combustion motors, recommended that for the purpose of the standard, air—assumed to be a perfect gas having a value of $\gamma = 1.4$—should be taken as the working fluid. For the ordinary four-stroke cycle engine, the formula giving the efficiency then is

$$\gamma = 1 - \left(\frac{1}{r}\right)^{0.4};$$

where $\frac{1}{r}$ is the ratio of the minimum volume to maximum volume. The Committee were satisfied that with good engines, giving their best economy, the actual efficiency divided by the ideal efficiency determined by this standard could be expressed by a ratio which varied between 0.5 and 0.7. This was deduced from separate tests made by Professor Meyer and Professor Burstall. Professor Burstall's tests also showed how inefficient design would decrease the ratio, as in some of his tests means involving greatly increased cooling surfaces were employed to increase the compression, and were found to considerably diminish the ratio. These tests showed further how too high flame temperature also decreased the ratio. The Committee required, however, further knowledge as to the

effect of the dimensions of the engine on the ratio, and accordingly they made tests on three engines of 5-inch, 9-inch, and 14-inch-diameter cylinders respectively, giving 6, 24, 60 I.H.P. In these engines, taking the mechanical efficiency to be 88 per cent., and calculating the I.H.P. from B.H.P., they found that the efficiency ratios were 0.61, .65 and 0.69 in the three engines. The tests showed, therefore, that by bearing in mind the slight changes in the ratio due to difference in dimensions, a close approximation to the best indicated efficiency to be expected from a given compression could be obtained by the use of a factor varying between 0.60 and 0.70, according to the dimensions of the engine. The tests also showed very clearly the small increase in economy of large engines in comparison with small ones, there being only 12 per cent. increase between 6 horsepower and 60 horsepower. The possible efficiency with the actual fluid used in the engine was known to be less than that given by the air standard. The Committee considered that a definitely known standard from which the actual efficiency could be deduced by using a multiplier found experimentally, allowing for the imperfections of the engine as well as for variations in the properties of the working fluid, should be adopted until the properties of the working fluid were accurately known. The author has examined the results of the test made by the Committee, and has made some further experiments on the large engine used in the tests, with a view to finding the true heat distribution in the engine.

The balance sheet given by the committee is as follows:

	L	R	X
Exhaust waste, . . .	35.3	40.0	39.5
Jacket waste, . . .	23.5	29.3	25.0
Radiation, ...	7.6	10.0	7.3
B.H.P., ...	26.7	28.3	29.8
	93.1	107.6	101.6

In obtaining this balance sheet the exhaust waste was determined by calorimeter, jacket waste measured, and the

radiation includes friction of the working parts. The B.H.P. was determined by rope brake. In order to reason as regards properties of the working fluid, it is necessary to know the I.H.P., the loss of heat during explosion and expansion, and the heat in the gases at the end of expansion. These quantities are not given in the ordinary balance sheet, as determined above. In the ordinary test the jacket loss is always over-estimated, because some heat which ought to go to the exhaust calorimeter flows to the water jacket after the opening of the exhaust valve and all through the exhaust stroke of the engine. The piston friction also will appear in the water jacket. The author has therefore attempted to adjust the balance sheet from data given in the Committee's report. Taking the mechanical efficiencies for the three engines, L, R and X, as 0.84, 0.85 and 0.86, the friction percentage of total heat is 5.1, 5 and 4.9, respectively. Deducting this from the jacket waste, corrected values for heat to water jacket, 21, 26.8 and 22.6 per cent. are obtained. Using these values, and reducing to percentage, assuming that the error in total heat is not in the I.H.P. item, a new balance sheet is obtained:

	L	R	X
Exhaust waste, ...	41.1	37.1	39.9
Jacket waste, True radiation, ...	27.1	29.6	25.4
I.H.P., ...	31.8	33.3	34.7
	100.0	100.0	100.0

The ideal efficiencies in these engines are practically the same, and assuming that one-third of the heat going to the engine is converted to work, and that the heat loss occurs near the beginning of the stroke, the difference between the jacket plus radiation losses in any two engines should be three times the difference between the I.H.P.'s. In the L and X engines this is found to be exactly the case. The jacket waste in the L engine is evidently too low, and on the above considerations should be 34.1. This value of jacket waste for the L engine will give an exhaust waste of 34.1, which is

practically the same as that determined by calorimeter. It appears, therefore, that in the L engine some heat which should have appeared in the water jacket has been lost. This corrected balance sheet is probably more accurate than that obtained in the test; but there is still some heat found in the water jacket which should be in the exhaust. The experiments give no means of determining this amount. The balance sheet, however, gives a method of calculating the maximum possible efficiency of the actual fluid. Adding exhaust waste to I.HP., and dividing I.H.P. by the sum, possible efficiencies for the three engines of 0.482, 0.473 and 0.465 are obtained. In obtaining these efficiency values, however, it has been assumed that the heat is lost at the beginning of the stroke, and, therefore, the values are not accurate. If the distribution of heat loss were known, the true adiabatic could be constructed and correct results obtained.

To check the results, indicator diagrams which give the correct mean pressure, have been studied. From the composition of the exhaust gases and the charge temperature, the weight of the charge is found to be 0.14 pound. From the diagram, the temperature drop from the end of expansion to charge temperature is 1,745 degrees Fahrenheit. The specific heat of the gases by weight, assumed constant, is 0.185. From these values, obtained from the numbers given in the Committee's report, it appears that 43 per cent. of all the heat of the combustible gases is accounted for in the exhaust. This gives a balance sheet for the X engine:

	Per cent.
Exhaust waste, ...	43.0
Jacket waste and radiation, ...	22.3
I.H.P., ...	34.7

The exhaust waste here would obviously be greater if specific heat increases with temperature. From this balance sheet, calculated as before, an efficiency of 0.447 is obtained; with air the efficiency would be 0.49. These considerations show the difficulty in using the actual fluid as a standard. In

spite of the great labor expended on experiments, only a rough approximation to the true heat distribution can be arrived at.

In 1884 the author made experiments on cooling after explosion in a closed vessel. Many other investigators have since done similar work, but cooling of a cylinder having a moving piston had never been investigated. The author made further experiments, and determined the cooling in the X engine. The engine was run at normal speed, and when a charge had been drawn in, the rollers actuating the inlet and exhaust valves were slipped, so that the valves remained shut. The explosion then took place, and the gases instead of being discharged were alternately compressed and expanded. An indicator card gives a cooling curve, showing temperature fall during successive revolutions of the engine. From these cards the mean apparent specific heat of the gases in the cylinder has been deduced, the gases being practically the same composition as those in the Committee trials. The values given increase with the increase of temperature, and have been called apparent specific heat values, because certain facts discovered are inconsistent with the change, being entirely specific heat change. Calculations, assuming these numbers to be the true specific heats are, however, very nearly accurate. From the cooling curves and specific heat values so determined a balance sheet has been obtained for the X engine as follows:—

	Per cent.
Heat flow during explosion and expansion, . .	16.1
Heat contained in gases at end of expansion, . .	49.3
Indicated work, ...	34.6

Comparing this with that found by the Committee, it is seen that the indicated work is the same in both. There is, however, less heat flow during expansion, and more heat in the gases at exhaust. This shows that about 21 per cent. of the heat in the gases at the end of expansion goes to the water jacket during the opening of exhaust valve and exhaust stroke.

This is considered a more accurate balance sheet than has yet been obtained. Calculating the ideal efficiency as before, the value 41 per cent. is obtained. From the values of specific heat given, the adiabatic may be calculated, from which the ideal efficiency is found to be 39.5 per cent., showing that the actual engine has converted 88 per cent. of the heat which it possibly could convert into indicated work. The new method has been checked by a test of a small Stockport engine in the author's laboratory, which gave similar results to those given by the X engine.

Tables have been calculated showing the ideal efficiencies for different compressions using the specific heat values given, and showing that, roughly, the air standard is 20 per cent. too high, and that if gamma be taken 1.285 for the explosion line, and 1.37 for the compression line, the change of specific heat between 1,700 degrees Centigrade and 1,000 degrees Centigrade, commonly used in practice, is too small to produce much error. More investigation is, however, required before even the apparent specific heat values can be accurately known for the various mixtures used in internal-combustion motors.

Much has been recently done, including experiments by Professor Hopkinson, Messrs. Bairstow and Alexander and Professor Burstall; but until further knowledge is obtained the air standard as defined by the Committee gives the best basis for comparing the performances of different engines.

The appendixes show the method of calculating the suction temperature, charge temperature, exhaust temperature and charge weight, and also method of calculating the adiabatic and efficiency for varying specific heats.—"The Engineer."

THE CRUISER OF THE FUTURE.

The three new cruisers of the *Invincible* class have been launched. The first, the *Indomitable*, was floated from the Fairfield Works at Glasgow; the *Inflexible* from the Clydebank yard of Messrs. John Brown & Co., Limited; and the *Invincible* from the Elswick Works of Sir W. G. Armstrong,

Whitworth & Co., Limited. These vessels are the most remarkable cruisers yet built, alike for their gun power and speed, and may be accepted as suggestive of the Admiralty's conception of the cruiser of the future. Their design has aroused considerable interest and criticism, not alone because of their great cost, which averages 1,729,000 pounds each—equal to the price of a modern battleship—but because of a sharp difference of opinion as to naval tactics applicable to cruisers.

It is being more fully realized than formerly that even the principles of strategy must vary or progress with the change of weapons. The enormous advantage of wireless telegraphy, if not also of torpedo and submarine craft, must influence the practice of war. For instance, as Mr. Julian Corbett pointed out in his lecture last week at the Royal United Service Institution, a close blockade has now become impossible, owing to the advent of these three important auxiliaries in naval warfare. In carrying out an open blockade or investment of a port so that the watching ships shall be beyond the range of torpedo action either from surface or submarine craft, it becomes necessary to have an exceptionally high speed. Only thus may full advantage be taken of geographical conditions—*i. e.*, of temporary lairs, from whence to make a dash after the enemy in the event of his trying to escape. The size of the area within which such positions may be sought is in direct proportion to the speed. If strategy may thus be influenced by advances in connection with the improvements in *matériel*, tactics are to be more extensively affected, and this is particularly so in connection with cruisers. Wireless telegraphy has enormously increased the possible area within a screen formed by the scouts of a squadron. As a consequence, it is impossible to ascertain the full strength of an opposing squadron within such a screen without driving home a reconnaissance in force. This necessitates powerful cruisers, so that they may come within sight of the main force of the enemy without suffering. Accurate information must be got at all costs. Unless this risk can be taken,

no admiral can be certain of the adequacy of his scouting force. The futility of depending on cruisers of deficient power, when the enemy may use as scouts the fastest of his well-gunned ships, is fully recognized. These vessels of the *Invincible* class have consequently been made the most powerful cruisers yet conceived, so far as present knowledge goes.

Indeed, these vessels more closely resemble battleships than any cruiser yet built. Their armor is only excelled by the ships of the *King Edward* class and the *Dreadnought* class. Their armament is equalled in broadside fire, or in bow and stern fire, only by the vessels of the *Dreadnought* class. Their speed of 25 knots is immensely superior to that of any armored ship afloat. As regards coal capacity, they are equal to any cruiser in the service, carrying 1,000 tons on normal draught; and although the high power necessary to give the maximum speed will make severe demands on the fuel supply, the radius of action should be satisfactory. This is the element in design which is most likely to be criticized on the ground that speed has probably been bought at some cost in respect of endurance. This raises the interesting question as to whether speed or endurance is the preferable quality in cruisers. It has been a cardinal principle in British warship design that radius of action is of greater consequence to a British fleet than to a foreign fleet, but there is suggestion now of some tendency to modification of this view. For over-sea work endurance is more essential, perhaps, than speed; but in the "narrow seas" speed is probably preferable. The number and distribution of our coaling stations greatly simplifies the problem so far as our Navy is concerned. On the other hand, the absence of such coaling stations intensifies the difficulties experienced by foreign powers. In the competitive game the more we can "force our rivals to sacrifice endurance to speed, the easier is our Imperial defense." Herein lies the strategical dilemma of the foreign naval designer, the one which has involved serious delay in the laying down of ships to meet our *Dreadnoughts* and our *Invincibles*.

The problem of the protection of merchant shipping may somewhat complicate the situation. The Admiralty view,

which must be accepted as the result of careful analysis of war experience, and therefore most reliable, is that the most effective procedure is to engage the enemy before the dissipation of his forces for the attack of commerce. Such dissipation of force, however, would be exceedingly dangerous, because the influence of wireless telegraphy must soon result in the attacking ship being mastered by a cruiser squadron, presuming even that she could keep the seas without returning for supplies to ports more or less closely invested by our fleets. There are those, however, who consider that special commerce protectors should be provided. Indeed, they contend in favor of the maintenance for the purpose of ships more or less obsolescent, and this view is based on the success of Japanese ships inferior to some of our vessels now regarded as practically obsolete. There is, as we have time and again pointed out, a danger of deducing lessons from the Russo-Japanese war which are not justified, owing to the ineptitude of the Russian ships or the special conditions prevailing. To utilize experienced officers and men in slow or defectively-armed ships, which would be at the mercy of one modern high-speed cruiser with high-velocity guns, seems to be courting disaster. The present trend is distinctly in favor of cruisers acting in squadrons in order to bring the enemy to fleet action, and to make it impossible, if not dangerous, to engage in a *guerre de course.* As Sir Charles Dilke pointed out at the London Chamber of Commerce on Wednesday evening, a neighboring naval power, after an inquiry by most competent authorities, had abandoned the view that it could, while not holding the seas against us, inflict great damage to our commerce.

The combination of qualities in the new cruisers has necessitated an exceptionally long ship. Hitherto the longest of the British modern cruisers have been the *Powerful* and *Terrible*, and the four armored ships of the *Drake* class, all of which had a length of 500 feet and a beam of 71 feet. Following the *Drake* class, completed six years ago, there was a tendency to decrease the length of the vessels, but later there has been steady advance. Thus the County class are 440 feet, the *Devonshire* class 450 feet, the *Duke of Edinburghs*

480 feet, and the *Minotaurs* 490 feet. The beam has, in the same period, been increased from 66 feet to 74½ feet. The three *Invincible* cruisers are 530 feet long and 78 feet 6 inches beam, the draught continuing, as in most of the recent armored cruisers, at 26 feet. This increase in length of 30 feet, as compared with the longest preceding cruiser, and of 40 feet as compared with the immediate predecessor, is necessary partly to enable high speed to be realized with the minimum of power, as the form for a given displacement may be finer, but it is a consequence also of the need for satisfactory disposition of the guns. It was laid down as an essential condition, first, that all the guns of the primary battery should be of 12-inch caliber—a condition never before exacted in any ship except the *Dreadnought*—and, second, that all the guns should have an arc of training to enable them to fire on either broadside and through very large arcs forward and aft. Eight guns are mounted in pairs in barbettes, one forward and one aft, and one on each broadside, but not, as in previous ships, on the same transverse line, the port barbette being some distance forward of the starboard barbette. This constitutes, as we have already indicated, a broadside armament equal to that of any ship afloat, equal even to the *Dreadnought*, and it will be interesting to show in tabular form the progress in the gun-power of successive armored cruisers.

BROADSIDE FIRE OF SUCCESSIVE ARMORED CRUISERS.

Class and year of launch.	Designed speed.	Displacement.	Number and caliber of guns firing on each broadside.	Collective muzzle energy from one round.
	knots.	*tons.*		*foot-tons.*
County (1900)	23	9,800	Nine 6-in.	30,200
Drake (1901)	23	14,100	Two 9.2-in. Eight 6-in.	63,600
Devonshire (1904)	22.25	10,850	Three 7.5-in. Three 6-in.	40,400
Duke of Edinburgh (1904)	22.75	13,550	Four 9.2-in. Five 6-in.	99,500
Minotaur (1906)	23	14,600	Four 9.2-in. Five 7.5-in.	137,000
Invincible (1907)	25	17,250	Eight 12-in.	381,576

It will be seen that in seven years the displacement tonnage has been doubled, and that the collective muzzle energy from one round of guns has increased more than twelvefold. What is of more importance, however, is that the County class are only capable of fighting at three miles' range, and even then against inferior ships. It is true that the *Drake* class of six years ago can effectively bring to bear two guns at four miles' range; the *Duke of Edinburgh* class and the *Minotaur* class, four guns of the same range; whereas the *Invincibles* will be able to utilize all of their eight guns at five miles' range, and then to do effective work against any foreign battleship. In this way they will be able to combat an equal number of battleships of the enemy's force while doing reconnaissance work, having at the same time a speed which will enable them, after gleaning all information as to the force of the enemy, to return to the admiral with full knowledge of the strength of his opposing force. Although their armor protection may not be as effective as the latest of our battleships, they will still be able to take their place in the line of battle; and on the principle that the most effective defence is an active and preponderating offence, they will do effective duty.

The value of the superior gun power is further considerably augmented, not only by the unification of the caliber of the gun for reasons which were admirably enunciated by Lieut. Comdr. Sims, but because of the exceptionally high freeboard of the ships. In the preceding cruisers—notably the *Duke of Edinburgh* class and the *Minotaur*—the forward guns were placed on the forecastle, which was cut away on each side to enable the wing guns on the upper-deck level to fire ahead. These wing guns, like all the other primary weapons in the ship, are in these earlier cruisers on the upper-deck level. In the new ships the broadside guns are on the same level as the forecastle, and therefore at a great height above the water line. The aft pair of guns alone are on the upper-deck level; but the deck erection to the rear of these stern guns is cut away at an acute angle on each side to enable the aft weapons to train as far as possible forward of the beam.

The disposition of the guns and machinery has involved difficult problems in order to secure a wide arc of training, and as this is a matter of increasing importance in the design of warships, considerable interest will be taken in the paper to be read next Wednesday at the institution of Naval Architects by Mr. James McKechnie, of Barrow-in-Furness, as his unique experience of both propelling and gun machinery entitles him to speak with authority on "the influence of machinery on the gun power of the modern warship."

As to the armor protection, this, as in the earlier ships, is continued right fore and aft, and extends from a considerable distance below the water line to the upper-deck level, the gun-machinery above this being further protected by the barbettes. The armor on the broadside amidships is, for some distance, 7 inches in thickness, reduced forward and aft to 4 inches, while all of the guns and the machinery are thus within heavy armor. Although the protection is not equal to that of the *Dreadnought*, it corresponds with the plating of the *Duncan* class, with this further advantage, that it is continued right aft as well as forward. As suggestive of the relative extent of armor protection in modern cruisers, it may be added that the proportion of weight of hull to the displacement tonnage of the ship in the County class was 60 per cent., in the *Devonshire* class 61.5 per cent., in the *Duke of Edinburgh* class 59 per cent., in the *Minotaur* 56.3 per cent., and in the *Invincible* 56 per cent. The inference may be justified that the weight devoted to protection is relatively less than in the earlier ships; but the length of the ship is a factor in such comparison, and in any case the difference is small.

The machinery is of the Parsons type, with water-tube boilers designed to give 41,000 horsepower. The disposition of the machinery has been arranged to enable a magazine for the amidship guns to be placed under the guns, and there are three boiler compartments, with two engine compartments, the latter divided by a longitudinal bulkhead. The turbines are arranged as in the *Dreadnought*. There are four shafts. On each of the center shafts there are fitted a cruising turbine

and low-pressure ahead and astern turbines, the two latter being within one casing. On each of the wing shafts there are a high-pressure ahead and high-pressure astern turbine. At cruising speed, therefore, the sequence, going ahead, will be, the cruising turbine on the inner shaft, the high-pressure turbine on the outer shaft, and the low-pressure turbine on the inner shaft; while for going astern the sequence will be, the high-pressure turbine on the outer shaft, and a low-pressure turbine on the inner shaft. There are four propellers, one on each shaft, and the outer propellers are about 20 feet ahead of the inner. The two inner shafts are carried within spectacle framing and stern bracket, and are immediately ahead of the two rudders. These, as in the *Dreadnought*, hang from the stern structure in a coned bearing, and they are further secured by nuts. These rudders are of very ample area, so that the turning power of the ship will be considerable.—"Engineering," March 15, 1907.

MARINE GAS PROPULSION.

THE DIFFICULTIES IN THE WAY OF APPLYING THE GAS ENGINE TO THE PROPULSION OF LARGE SHIPS AND A POSSIBLE SOLUTION.

By A. VENELL COSTER, Manchester Association of Engineers.

Last month we reviewed in these pages an article dealing with the possibilities of the internal-combustion engine for marine purposes which considered only the mechanism and arrangement of the engine. A paper on the same subject, by Mr. A. Vennell Coster, recently read before the Manchester Association of Engineers, contains a discussion of fuels and gas producers and suggests an arrangement of engines which differs widely from that proposed in the former article. A few extracts are taken from Mr. Coster's paper with a view to showing how he proposes to solve the difficulties in the way of applying the gas engine to the propulsion of large ships.

"I feel confident that if a marine engineer were told that by the introduction of the gas installation he would obtain the fol-

lowing advantages, he would be more than satisfied with the substitution of gas for steam power.

"(1) The ship driven with half the amount of fuel.

"(2) Stand-by losses reduced over 75 per cent.

"(3) Working pressure confined to the engine cylinders.

"(4) No boiler tubes or main steam pipes to burst, or furnace crowns to collapse.

"(5) No priming in a heavy sea way, or water hammer in pipes and cylinders.

"(6) No more difficulties with the firing of boilers in a beam sea. Gas producers may be charged only twice every twenty-four hours, and the rolling and pitching of the vessel is rather an advantage than otherwise in assisting the fuel down from the charging hoppers.

"Firms of the standing of Messrs. Beardmore, Thornycroft, Yarrow, Crossley Bros., Nobel Bros., Sulzer Bros., Mr. H. Cherry and many others have already applied themselves to this venture, and the results so far have been by no means discouraging. There are still many difficulties to be faced and overcome, but they will be overcome. In this paper my proposal is to put before you such a scheme of marine gas propulsion, that the most cautious of us would be willing to risk a voyage by its means. What are the main difficulties?

"1st. The construction of a gas producer able to gasify all grades of bituminous coal.

"2d. A simple method to cleanse the gas from tar, either before the introduction of the fuel into the producer proper; when in the producer; or after the gas has left the producer on its way to the engine.

"3d. Perfect control of the gas-propelled vessel in starting, stopping, reversing and running at all speeds.

"These are the three main difficulties in our way.

"Dealing with No. 1 and No. 2. If anthracite coal or coke is used these difficulties are entirely avoided, but with a limited supply of anthracite our wisest course is to leave it out of the question altogether. For marine purposes we must deal with the ordinary coal that may be obtained, not only in Wales, but

in various parts of the United Kingdom, and the many coal-fields scattered about the world. Bituminous coal is cheap compared with either anthracite or coke, and as I have already stated, it is the only possible fuel for marine gas producers. Its thermal value varies from 10,000 to 14,000 British thermal units per pound, and costs from six shillings to seven shillings per ton. For many years gas engines have been driven by bituminous coal gas, and the difficulty has always been to get rid of the condensible hydro-carbons which would fill and clog the valves and gas passages of any engine they were allowed to enter. This problem is one that must be faced rather by our chemists than by our engineers, and many notable firms have already solved the problem, but not in that simple manner that would make the gas plant perfectly suitable for marine purposes. It is reported from several trustworthy sources that a simple solution has been discovered.

"Then there is the necessity of dealing with the many grades of bituminous coal, some suitable, many unsuitable. Shipowners would have to insist on a correct analysis being supplied prior to delivery, so as to ensure gas producing qualities.

"The third difficulty, namely: Perfect control of a gas-propelled vessel is our next consideration.

"One of the main difficulties that confronts the gas engineer in applying gas engines for marine purposes is that of reversing. With small powers it is possible to reverse by means of a reversible propeller, the engine having only one direction of rotation. There is naturally a very serious objection to putting the reversing gear in so inaccessible a position as in the propeller, but at the same time, there is undoubtedly a future for this type for river and coast work, if made simpler and more reliable. For higher powers up to say 500 horsepower units there is another system in vogue for reversing the propeller, namely—a reversing gear, which may be a combination of bevel or spur gear and friction clutches, is fitted somewhere on the propeller shaft within the vessel and in an accessible position. But for powers above 500 horsepower the gas engines themselves must be made to reverse, for this dispenses with

expensive and complicated clutches. There is no practical difficulty in reversing the gas engine, provided a sufficient reservoir of compressed air is at command.

"But the greatest of all difficulties in the application of gas engines for marine purposes, is the attainment of a sufficient range of mean effective pressures in the engine cylinders, so that like a steam engine, varying speeds, from full speed to dead slow, can be run without fear of stopping. We may reasonably assume that a maximum reduction of sixty per cent of the mean pressure leads only to a reduction of thirty-five per cent. of the speed of rotation, or, if half the cylinders are cut out, the speed of revolution will be reduced by about thirty-two per cent. Then by reducing the mean pressure in the remaining cylinders, the result will be a reduction of the effective turning effort to twenty-five per cent. of that of full power, the speed of revolution will be something less than half that of full power. This reduction of speed is not satisfactory for marine purposes and it is doubtful whether gas engines could be run with so great a reduction of explosive charge. As an example, the marine gas engine in the *Lord Antrim* (mentioned below), with normal speed of 150 revolutions, can not safely be reduced below seventy-five revolutions per minute, because with a light fly-wheel and possible miss-fires, the engine has not sufficient momentum to carry it over the compression strokes.

"The following solution at once meets all these difficulties, whether the vessel has to run full speed or dead slow, by a practical manipulation of the speeds of independent engine units. In running some of them ahead and some of them astern any speed in either direction can be readily attained by the officer in charge of the vessel on the bridge, and without stopping, except during reversal, a single engine or the normal flow of gas through the gas producers.

"By this means the author attempts a solution (analogous to the turbine problem), but on different lines from those apparently adopted by builders of marine gas engines of 500 horse-power and over. They rely on a large reservoir of compressed air for maneuvering and working the vessel up dock, in nar-

row waters, or when running dead slow in foggy weather, because with one engine unit, as has already been stated, the vessel can not be driven by gas much below half speed. But to depend on compressed air at these times, when engines and producers should prove not their inefficiency but efficiency, is to stultify the whole subject of marine gas propulsion."

Mr. Coster illustrates his solution by means of a concrete example. He takes the *Lord Antrim,* a vessel 375 feet long by forty-seven feet beam and twenty-eight feet depth. It is at present run by one triple-compound engine, running sixty-seven revolutions per minute and developing 2,360 indicated horsepower. For this engine Mr. Coster proposes to substitute three gas engines developing the same horsepower as the steam engine, but running at the increased speed of 150 revolutions per minute, the gas being supplied by four producers with their auxiliary apparatus, these producers having sufficient capacity to supply, besides the main engines, several smaller units for the driving of dynamos, electricity being used to drive the air-compressing apparatus, the pumps and other necessary machinery.

Each of the main engines runs a separate propeller shaft. "The central shaft is fitted with a larger propeller than the two wing shafts. These have two propellers carried on **A** frames from the bilge of the vessel, and do not project beyond the lines of the vessel, and are thus satisfactorily protected. It is possible, as already intimated, with this arrangement of machinery, to run the vessel at any speed from full speed to dead slow either ahead or astern, without stopping the engines except for reversing. For example: taking the propeller going astern as having only fifty per cent. efficiency when compared with going ahead, by running the center engine full speed ahead, and the wing propellers about full speed astern, the vessel can be practically stationary. Then by slowing down the wing propellers to half speed astern, the vessel can be made to move about quarter speed ahead, and as these engines can vary their speed from 150 revolutions down to seventy-five revolutions per minute, any speed in any direction can be attained by the vessel without stopping the engines. In narrow waters, en-

tering or leaving port, or working up dock, the vessel is under as complete control as if fitted with the latest steam engines, and with no fear of the quality of gas being reduced. With such an installation, which occupies somewhat less space, less weight, and about the same cost as a steam installation, but with about fifty per cent. less coal consumption, who would fear a deep-sea voyage if the machinery were in charge of experienced and capable marine gas engineers?

"There is no question in my mind as to what type of gas engine we should adopt for marine purposes, the simplest is the best, and there is no type so simple and satisfactory as the four-cycle or 'Otto' type, with its sequence of suction, compression, power, and exhaust strokes, taking place in each cylinder. This type of engine, with inverted vertical cylinders, has a mechanical efficiency of over ninety per cent. and a thermal efficiency of thirty three per cent. and for a vessel requiring say 3,000 horse-power units of six single-acting cylinders, with pistons of say twenty-one-inch diameter by twenty-four-inch stroke, would be suitable. But where larger powers are required, then tandem cylinder engines with open cylinder liners, having piston rod and stuffing box between the tandem cylinders, where they can be easily inspected and adjusted, would be required."—"The Engineering Magazine."

HOW COMPRESSED AIR RAISED A SUNKEN SHIP.

THE REMARKABLE SALVING OF THE STEAMSHIP *BAVARIAN*.

To turn a 12,000-ton steamship into a huge steel bubble by pumping her full of compressed air is a recent engineering feat.

The steamship *Bavarian* of the Allan Line, ran on Wye Rock, thirty-eight miles below Quebec, on the night of November 3, 1905, a few minutes before high tide. Almost every method known to wreckers for salving the vessel was tried and found wanting and over $150,000 was spent in these efforts.

Examination had shown that the *Bavarian's* bottom amidships was in a very ragged condition. The holes were so large that it would be hopeless to try to pump the water out. Preparations were accordingly made to treat the holds as caissons, compressed air being used to force the water out through the opening in the bottom.

Work was begun by the North American Wrecking Company on September 7, 1906, and the vessel was floated on November 16, 1906. Everything in regard to the operations was calculated with mathematical accuracy. The calculations for the buoyancy required, and at the points chosen, were most fortunate.

It was necessary to timber solidly between decks above the several compartments that were to be used as caissons. The hatches were closed by plating. Air locks were placed on the compartments which were to be treated as caissons. Every opening in the deck, scuppers, etc., was closed.

When the air was applied the water rapidly receded and workmen were able to stop the rents in the bottom with temporary plating. In some of the holds even the leaks were not closed, and the vessel was floated without a bottom. Pressure men, that remarkable class of men who make it their business to work in compressed air, and who are commonly known as "sand hogs," were brought from the Quebec Bridge, the caisson work of which had shortly been completed, or from New York, the superintendent of the work having for many years been engaged in compressed-air work about New York.

A wooden tank of about 200 tons capacity was built directly between the engines, and the weight of the engines (180 tons) was carried by this tank. On the day of flotation about twenty-five tones of water were left in the tank. As the vessel rose and the engines settled to their old level, blocking was put between the tank and the deck over it, and this water pumped out, the surplus lift of the tank being transferred to the vessel. The heavy tides of the St. Lawrence, although the center of the vessel was flooded, lifted the end of the vessel, and the craft rose and fell with high tide, so that the engines rose and fell on some occasions fourteen inches.

Air bags and tested barrels were used in the after bunkers. The boilers were blown out, and air was applied to the forward bunkers, they, too, being treated as caissons.

Several of the holds where the bottom was not so badly destroyed that it was necessary to treat them as caissons were pumped out, eight and ten-inch centrifugal pumps being used for this purpose.

Owing to bad weather the tugs which had been lying alongside on November 16, the date set for flotation, had dropped down the river to a more sheltered position. As the tide rose the air compressors were set to work and the full power of the plant used in forcing air into the holds of the ship. Suddenly there was a movement in the great vessel as she lifted herself from the rock and a cheer went up from those on board. Five minutes later the *Bavarian* floated clear of Wye Rock in sixty feet of water, and was hauled to her anchors, which had been set off her port bow and quarter. After the first few minutes all apprehension that the vessel might turn turtle or that the air pressure would not hold the water back was dispelled. The *Bavarian* floated on an almost even keel and was shortly after towed by tugs to Quebec. The wrecking operations were under charge of William Wallace Wotherspoon, C.E., superintendent. who had entire charge of the wrecking operations inside the bulwarks of the vessel, and Mr. R. O. King, C.E., who was controlling engineer. Capt. W. Leslie, of Kingston, had charge of all nautical work.—"Scientific American."

CONTROLLING TORPEDOES BY WIRELESS TELEGRAPHY.

A torpedo-launching apparatus, which is operated by electric waves, has recently been tried in France, on the Mediterranean coast. It consists of two cylinders placed one above the other, the upper one acting as a float which holds the mast wire, while the lower cylinder contains the torpedo and the launching device. The method of carrying out the maneuvers of the apparatus from a distance by the electric waves has been devised by M. Devaux, who gives the following account of the

method. Up to the present the Hertz waves have been utilized generally for operating an electro-magnet, whose armature makes the well-known signals as in the Morse instrument, and this is the method used in wireless telegraphy. However, the movement of the armature can evidently be used to set free the force of another instrument, thus acting as a relay. Up to the present, the devices in use allowed of using one electro-magnet for operating two movements only, corresponding to the up or down position of the armature. To carry out a series of maneuvers which have no connection with each other, it is necessary to have as many electro-magnets as there are operations. This, though complicated, would be possible in the case of wire transmission, but it is not practicable for electric waves, seeing that as yet we are unable to separate the waves so as to have them act on different coherers located at the same point. We proposed the problem of setting in movement from a distance by electric waves, of a series of forces, acting in an order always variable and keeping independent of each other, and we devised a new method for the operating apparatus. This apparatus is used with each of the coherers, and can also be used with a wire-transmission system. In the latter case they need only one wire, using the earth as a return.

The present system consists, first of a distributing device passing over all the contact points from which start the different operating circuits, and second, a commutating switch which sends the current at the right moment when the distributor has reached the proper circuit. To carry this out, an electro-

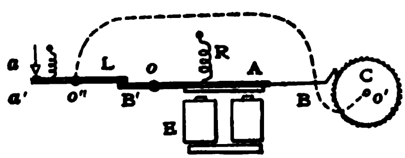


Fig. 1.

magnet *E* (Fig. 1) can attract an armature, *A*, which is held back by the spring, *R*, and is pivoted at *O*. This armature is prolonged at each end by the arms *B* and *B*.' The arm, *A*,

provided with a pawl, works upon the ratchet wheel, *C*, which is mounted on the shaft, *O′* of the distributor. The wheel thus advances one tooth at each attraction of the electro-magnet. At the other end, the arm *B′* strikes each time against the end of the lever, *L*, which is pivoted at *O″* and which works the contacts, *a a′*. Mounted on the shaft, *O′*, of the distributor is an arm, *D* (Fig. 2), whose end works upon a set of

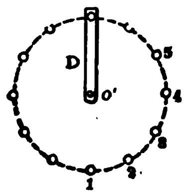


Fig. 2.

contacts 1, 2, 3, etc., and sends current from *O′* into twelve circuits. Each movement of the electro-magnet thus moves the arm around one contact. The electro-magnet is operated by aerial waves, using a coherer, or again by a line wire. In Fig. 3 is seen the second part of the device

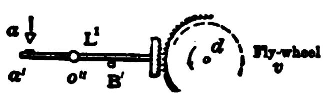


Fig. 3.

for shifting the circuits. The arm, *L′*, lies in another plane from the arm *B B′*, and on the forward end it carries a small rack and pinion device which is not seen in Fig. 1. The rack, sliding over the wheel, *d*, engages with it when the lever, *L*, is out of its normal position. In coming back, the rack draws the wheel with it, but as the latter carries a flywheel on the shaft, this gives it a certain inertia and retards the fall of the lever. During the series of attractions of the armature, the contact at *a a′* is thus kept open, and it is only when the armature is stopped and therefore when the distributor arm has come to the proper point, that the contact is closed.

The apparatus works as follows: When at rest, the arm, *D*,

remains upon contact number twelve, which is a dead point. The lever closes the circuit at *a a'*. We now wish to close circuit number seven without interfering with the others. It suffices to send seven currents or seven sets of waves with the rhythm $1 = t$. The electro-magnet will work seven times at this frequency and will make the ratchet wheel, *C*, advance by seven teeth, whereupon the arm, *D*, comes upon point number seven. But now the contact at *a a'* has remained open on account of the inertia, as we have seen, of wheel *d*, and it is only closed when the arm, *D*, comes to rest. We thus close circuit number seven without operating the others. Again, we may wish to close several circuits at the same time. To do this, the distributor must be free to move without breaking the circuit which has been closed. It suffices that circuits one, two, three...twelve be closed by locked relays, and the latter serve to operate the work circuits. These relays are closed and remain so when the current passes, and their opening depends upon a device connected with one of the contact blocks.

In Fig. 4 is shown the apparatus for operation at a distance. It is connected in the place of the usual Morse receiver for aerial telegraphy. We have applied this apparatus to operating

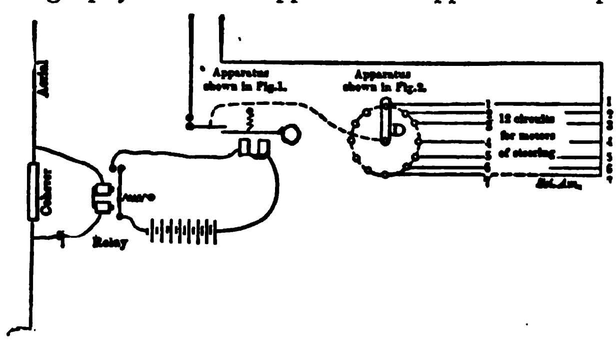


Fig. 4.

torpedoes at sea, and thus need the following maneuvers: 1. Forward run. 2. Back run. 3. Stop of propeller motor. 4. Rudder to right. 5. Rudder to left. 6. Stop of the steering motor. 7. Lighting of the signals at the front. 8. Lighting at rear. 9. Launching of the torpedo. The apparatus uses twelve points.

thus having three points free. The speed allowed for making a complete round of the distributor is two seconds. The nine working circuits closed upon seven locked relays which worked the propeller and steering motors, signals lamps and launching apparatus. Inside the torpedo is placed a set of Fulmen storage batteries which give a four-hours' run. The torpedo is formed of two cylinders of sheet iron with conical ends, connected one above the other. It weighs six and eight-tenths tons. The top cylinder, thirty feet long and two feet diameter, serves as a float, and it carries two small masts to which are fixed a receiving wire and a set of signal lamps. The lower cylinder, which is thirty-five feet long and three feet diameter, contains a torpedo-launching tube and a Whitehead torpedo, besides the battery, motors, etc. The land post has a mast wire fifty feet long, in fine strands. A distance from 400 to 5,000 feet was covered in the trial. The latter took place in the port of Antibes on the Mediterranean. These tests have proved satisfactory as regards the different maneuvers of the torpedo-launching apparatus from the post on shore.—"Scientific American."

HIGH DUTY METAL.

It is a well known fact that the copper alloys known as brass and bronze generally suffer a marked reduction in strength with increase of temperature. This loss of strength is not generally taken into account because the deterioration is slow up to a certain point, and this point is above the temperature of steam or air at the usual pressures. At about the temperature of 150 to 175 pounds steam the loss begins to be very rapid, and at about 400 degrees Fahrenheit there is a sudden marked drop in tenacity. From that point to 500 degrees Fahrenheit the loss is very rapid and the strength of the material is seriously affected.

In designing a new line of extra heavy and medium pressure brass valves, it was determined to use the metal which would show the smallest percentage of loss in this

respect that it was possible to obtain, the same metal to have other qualities which are essential in a valve at all times and particularly in a valve for use under high pressure.

In endeavoring to secure this metal, an effort was made through the Secretary of the American Society of Mechanical Engineers to collect all of the existing literature upon the subject, and it was discovered that practically all that has been determined on this question in the results of the experiments of the British Admiralty in 1877 is described on page 309 of Kent's Manual. Having found that no further light was to be obtained from past researches, there was inaugurated an exhaustive series of tests with a large number of mixtures of copper, tin, zinc and other metals, with the purpose of determining, absolutely, the best metal for the purpose.

These experiments have extended through a period of eight months, and the statements made above have been fully corroborated and emphasized. For instance, it was found that an alloy that is very commonly in use as steam metal and would be called a fairly good metal for this purpose, showed a drop in tensile strength of as much as 28 per cent. when raised to the temperature of 407 degrees Fahrenheit, which is about the temperature of steam under 250 pounds pressure without superheating. It was assumed that this temperature would be a fair one at which to make these tests, inasmuch as engineering practice is tending strongly in these days to the use of steam at about this pressure, or superheated from a lower pressure to this temperature.

Another metal which has been considered an excellent mixture and is frequently used by valve makers for valves of higher grade and designed for higher pressure, showed twenty-two per cent. loss under the same conditions.

The well known "government" mixture, as it has been called, consisting of eighty-eight parts of copper, ten of tin and two of zinc, was found to be as little affected by this extraordinary increase of temperature as any alloy which has ever been used, as far as known, in the manufacture of valves. The "government" mixture was found to have as an average of a large

number of bars tested, a cold tensile strength of 33,633 pounds per square inch. When raised to 407 degrees Fahrenheit and again tested, the temperature being maintained constant during the test, the tensile strength dropped to 30,675 pounds per square inch, showing a loss of nearly nine per cent.

We are able to say that, after making all of our experiments, we have arrived at an alloy which is practically of the same tensile strength as the above mixture when cold, as it shows an ultimate strength of 33,520 pounds per square inch at seventy degrees Fahrenheit, and further, it shows an ultimate strength of 31,627 pounds per square inch at 407 degrees Fahrenheit, the loss being only five and six-tenths per cent.

A table is appended which shows in summarized form the results of these experiments upon six different alloys, as follows:

	Tensile strength at 70° F.	Tensile Strength at 407° F.	Loss per cent.
No. 1	21,790	15,640	28.2
No. 2	29,010	22,410	22.4
No. 3	24,510	22,059	9.5
No. 4	33,633	30,675	8.8
No. 5	33,710	31,305	7.1
No. 6	33,520	31,627	5.6

Number one is the steam metal alluded to above and which is in common use among valve manufacturers.

Number two is a metal which we ourselves use at times and is fairly strong and durable at comparatively low temperatures.

Number three represents one of the earlier experiments illustrating the advance along the line of research, but indicates too low tensile strength.

Number four is the "government" mixture, so called.

Number five is one of the later attempts, and number six is the mixture which has finally been adopted and will be called by us "High Duty" metal.

In addition to the rare quality of maintaining the high tensile strength at high temperatures, this metal also shows

wearing qualities which are very remarkable; and as wearing qualities in the moving parts of a valve are of the utmost importance, the "government" mixture was again taken as the unit or standard, and it was found, by means of a special machine in which the movements of the parts of a valve were imitated exactly under extreme conditions of friction and at high temperature, that this metal wore away at about one-third the rate shown by the "government" mixture mentioned above.

This alloy has also been found to be very tough, resisting shock or water hammer with greater success, and is, in fact, far less brittle than any other of the metals tested.

In strength under compression it again showed marked superiority, as the flow of the metal was extremely uniform under high pressures, much more so than any other alloy upon which these tests were made.

Further, and this is very important, this metal makes sound, tight castings. The tests prove this conclusively.

WESTERN TUBE CO.

THE PRINCIPLES OF SIMILITUDE.

Some thirty years ago, or more, the late Professor James Thomson—who, in his own line, was as original and profound a thinker as his more famous, but younger, brother, Lord Kelvin—contributed to the Proceedings of the Institution of Engineers and Shipbuilders of Scotland a paper on the "Comparison of Similar Structures as to Elasticity, Strength and Stability." In this paper a number of very simple, but important, relations were established between the relative security of two structures differing from each other only in the matter of scale. Some eight years ago (see "Engineering," vol. lxviii., page 475) Professor Barr discussed the same subject with considerable wealth of illustration, and finally, in a paper read lately before the Institution of Automobile Engineers, 1, Albemarle Street, W., Mr. F. W. Lanchester, whose originality and success as a builder of motor cars of the highest class is well established, set forth once more these "Laws of Simili-

tude." In all these papers it has been shown that a model if made of the same material as its prototype will carry with the same factor of safety an equal load per square foot, if similarly distributed. At the same time the weight of the model varies obviously as the cube of the ratio of the scales, so that, so far as the stresses in it arise from the weight of its constituent parts, it is much less severely stressed. It is, therefore, quite safe to build a bridge or a roof by scaling down from a larger one designed to carry the same load per square foot. The stresses arising from the traveling load, or from wind pressure, will be identical in the reduced model, whilst the dead-load stresses will be less. Where, however, questions of stability rather than of strength are concerned, this process of scaling down is not free from danger, since the stability of a chimney, for instance, against overturning from a given wind-pressure per square foot is double that of another built to half scale. Hence, if Nature worked on strictly mathematical lines, a little woman should have disproportionately big feet as compared with her taller sisters.

The same theory of similitude shows that in a model engine the stresses which arise from a given steam pressure are exactly the same as in the larger original; and, further, that, so far as the inertia stresses are concerned, these will be the same in the two, if the piston speeds are the same. It follows, therefore, that the model can be run safely at a higher rate of revolution than the larger engine. The horsepower, developed is obviously proportional to the volume of steam passed per minute; that is to say, to $n\ d^2 \ . \ s$, where n denotes the number of revolutions per minute, d the diameter of cylinder, and s the stroke. Using the small letters for the model, and large ones for the full-sized engine, we have, if the piston speed is the same, $n\ s = N\ S$, so that the horsepower of the model varies as $N\ S\ .\ d^2$, whilst its weight will vary as d^3. Hence the horsepower per unit of weight will, in the case of the model, be proportional to $\frac{N\ S}{d}$, whilst that of the larger engine will vary

as $\frac{N S}{D}$; and as D is greater than d, much more power could obviously be obtained from a given weight of metal by using it to build a number of small motors than to construct one large one. In fact, the weight of the smaller engine per horsepower developed should vary directly as the scale of reduction. No doubt it is to this fact that a portion of the enormous saving in weight shown in Mr. McKechnie's recent paper before the Institute of Naval Architects, to follow on the substitution of gas for steam in the propulsion of war vessels, is to be attributed. He proposed, it will be remembered, the construction of a cruiser, in which the 16,000 horsepower required to drive her were developed by gas engines, with forty cylinders in all, driving four shafts. With steam this power would be developed by, say, two four-cylinder engines driving two shafts; and were the gas engines arranged in similar fashion, the principle of similitude shows that their weight would be increased in the ratio $\sqrt{\frac{40}{8}}$, or be more than doubled. The weight of the producers would not, of course, be affected, but the modification here suggested would undoubtedly go far to cancel a large proportion of the great gain in weight claimed.

This conclusion from the principle of similitude may of course be challenged on the perfectly correct contention that in practice the law in question is more commonly violated than confirmed, and it is no doubt for this reason that engineers have made so little use of it. The late Mr. Normand was, however, a firm believer in both its validity and its utility, and on several occasions constructed marine engines of different sizes from the same set of drawings, merely using a different scale. In one instance (see "Engineering, Vol. LX, page 58) the ratio of the scales of the two engines was as much as one to one and nine-tenths. If, however, comparison is made of the weights and outputs of motors of different kinds made in a large range of standard sizes by good firms, not the slightest sign of any agreement between the horsepower actually developed per unit of weight in the different sizes and that theoret-

ically due is in general visible. Thus, from the catalogue of a firm making a popular type of high-speed engines in sizes ranging from 250 up to about 1,700 horsepower, the ratio of dimensions and the weights per horsepower corrected to the same piston speed are as follow:

Ratio of cylinder diameters..........	0.42	0.54	0.66	0.80	1
Total weight per B.H.P., pounds..	65	66	64	62	57

Again, from the catalogue of a large turbine maker we get the following figures:

Ratio of inlet diameters..	0.092	0.202	0.356	0.536	0.735	1.0
Weight in pounds per B.H.P. at 16-ft. head...	52.0	31.5	36.2	42.0	38.1	52.9

So that in both cases the weights per brake horsepower hour, instead of being less in the smaller sizes in the ratio of the corresponding dimensions, as theory indicates they should be, are actually more or less constant. Quite similar results would be obtained on making a similar comparison for a series of gas engines of the same type made by a single firm.

The reasons for this extraordinary discrepancy between theory and actual fact appear to be mainly that a firm's standard practice as to the minimum thicknesses of castings and the like is based upon their heaviest and most important work. In getting out a set of standard proportions for a series of engines the basis is the largest engine of the group; but whilst, as has been shown, the smaller sizes would be amply strong enough if simply scaled down from the larger drawings, this is practically never done, but the minimum scantlings used are not vastly different throughout the whole range.

It follows, therefore, that a small engine designed by a draughtsman accustomed to heavy work is unduly massive, though it generally proves reliable and satisfactory in working. The saving of a few pounds of cheap metal in a small engine is not felt to be worth the trouble of training the foundry to produce light castings, or the mental worry of departing from established practice in minimum dimensions. The difficulty a draughtsman of this kind has in "scaling down" is best ex-

hibited perhaps when he is set to tackle instrument design. The earliest of the cyclometers put on the market, for example, was designed by a capable mechanical draughtsman, who made a thoroughly "substantial engineering job" of it, with the result that the weight of the instrument ran into pounds, and its cost was something between a guinea and thirty shillings. The modern cyclometers, on the other hand, are the product of experience in instrument work. They are quite reliable, though their weight is a mere ounce or so, and their cost in pence much the same as that of their predecessor was when expressed in shillings. Of course, the scaling-down process advocated by Mr. Normand requires to be applied with discretion. Sound castings cannot be obtained below certain minimum thicknesses, and if scaling down leads to less scantlings than these, the design must be correspondingly modified. Moreover, all structures and machines have to be capable of withstanding, in addition to their intended stresses, the effect of accidental shocks and blows. These, in the case of large engines, may be insignificant compared with the regular working loads; whilst in a much reduced model they may give rise to stresses and strains much in excess of those legitimately obtaining.

Owing to the unconscious bias of the designer to base his minimum dimensions on the heaviest class of work with which he has experience, the figures tabulated above cannot be considered as proof that the principles of similitude are inapplicable in practice. As already stated, Mr. Normand utilized them boldly, and if instead of comparing together the different sized productions of one firm, those of different firms, one of which specializes in heavy work and the other in smaller engines, are contrasted, there appears to be a very fair agreement between the theory and actual practice. From the paper on "Recent Trials of Machinery of Warships," contributed to the Proceedings of the Institution of Civil Engineers by Sir A. J. Durston, K.C.B., and Mr. J. Oram, we take the following particulars as to the engines of the *Powerful* (cruiser) and the *Bullfinch* (torpedo-boat destroyer):

......	Diameter of low-pressure cylinder.	Piston speed per minute	Steam pressure at engines.	Maximum horse-power.	Weight. / H.P.
	inches.	*feet.*	*pounds.*		
Powerful....................	76	906	207	25,900	93
Bullfinch....................	34	1,200	210	5,800	22

Reduced to the same piston speed, the weight of the *Powerful's* engines would be 77.5 pounds per horsepower, so that the ratio between the specific weights is 3.15; whilst if the materials and design were the same in the two cases, this ratio should by the theory be that of the low-pressure cylinders, or 2.24. In this case, therefore, part of the large reduction in the specific weight is undoubtedly due to the difference in the design and the materials.

Similarly, if, instead of comparing the specific weights of a range of gas engines constructed by one maker, the specific weight of his largest, and presumably most carefully proportioned, engine is compared with a motor-car engine, the difference is extraordinary. Here again the matter is complicated by differences in design and in the nature of the materials used; but the economy of weight shown is far more than can thus be accounted for. The effect of an improvement in the quality of materials in augmenting the horsepower obtainable from a cylinder of given capacity was discussed by Mr. Lanchester in his paper already referred to. He showed, for example, that a ten per cent. increase in the tensile strength of the metal used for the reciprocating parts of a given engine would allow of an increase of fifteen per cent. in the power developed. He also generalized the usual statement that the horsepower of similar engines running at the same piston speed should vary as the square of the linear dimensions by showing that a more complete statement of the facts of the case is that it should vary as $D^n S^{2-n}$, where D is the diameter of the cylinder, and S the stroke. By then investigating the relation of the reciprocating stresses to the diameter and stroke of the cylinder, he arrived at the conclusion that $n=1.5$, and therefore proposed that in

rating petrol engines in hill-climbing contests and the like, the power should be taken as equal to $0.4.D^{1.5}S^{0.5}$. With this formula, he contended, extravagant freak designs, built merely to "beat the rating," will be eliminated.

The laws of similitude, as set forth above, are incomplete in the regard that no allowance has been made for the fact that, in general, a large engine is somewhat more efficient than a small one. In the case of a steam engine, for instance, the initial condensation is increased the larger the proportion of the surface exposed to the volume of the clearances; and, consequently, to obtain the same mean pressure in a model as in a larger engine, a greater quantity of steam must be passed into the cylinders, so that if the port area bears the same proportion to cylinder area in the small as in the large sizes, there will be a greater loss from wire drawing, and to obtain the same mean pressure a little later cut-off will be required. In the case of internal-combustion engines the loss to the walls is similarly proportionately greater in small than in large engines, since the ratio of surface to volume increases as the scale of the machine is diminished. The mean effective pressure in such engines is therefore somewhat less in small than in large gas engines, though a certain compensation is possible by the adoption of higher compressions and relatively longer strokes in the smaller sizes. The Crossley engine at the suction gas trials at Derby, for instance, had a cylinder 8½ inches in diameter and a 21-inch stroke, with a compression of 210 pounds per square inch. Large engines are not built with anything like this ratio of stroke to diameter, nor to work with such a high compression; about 140 pounds being the maximum with which these large engines can be worked on a commercial basis with producer gas, though the attempt has frequently been made to increase this figure. These considerations therefore place a limit to the increase in the efficiency of a gas engine with increase in size. Most makers agree that a 60-horsepower gas engine will be practically as economical of fuel as a larger one. A complete theory of similitude

should take into account the whole of these factors, but the requisite data for effecting this in a thoroughly rational manner are as yet wanting.—"Engineering," May 10, 1907.

JAPANESE GOVERNMENT STEEL WORKS.

We have, from time to time, noted the progress which was being made with the Imperial Japanese Government Steel Works at Wakamatsu, as the matter had special interest to the steel makers of this country. That progress has been interrupted on account both of financial and technical difficulties, and before the war with Russia the prospects of the undertaking were not by any means bright. A national struggle, such as that in which Japan was engaged, gives an impetus to all the industrial undertakings which are necessary for carrying it on; and the Government Steel Works seem to have been helped over their chief difficulties, and are now in a position to do very effective work. A very complete report of their present condition has been received by the Board of Trade from H.M. Chargé d'Affaires at Tokio, which contains some points of interest. After sketching the history of the undertaking, the most of which we have noted on previous occasions, the writer of the report reviews the expenditure, and makes out that the total of the sums appropriated for the establishment of the works amounted to nearly £2,000,000.

The area of the works is about 330 acres, including some 82 acres of ground recently purchased for the purpose of enlargement and not yet built upon. The works are quite close to Wakamatsu, the chief port for the export of Kyushu coal, and about nine miles west of Moji, the well-known coaling port on the Shimonoseki Straits, and northern terminus of the Kyushu Railway. The position was chosen largely on account of its proximity to the Chikuho coal fields, by far the most extensive coal-producing district at present known in Japan. This district lies some thirty miles to the south of Wakamatsu, in the provinces of Chikuzen and Buzen, and covers an area of over 320 square miles. The entrance to Wakamatsu Harbor is very narrow, opening to a basin about

a mile across at its widest part. This basin, again, opens to a large lagoon some ten miles in circumference. It is on the eastern side of this lagoon that the Imperial Steel Works stand. The lagoon is naturally very shallow, but dredging operations have been carried on since 1890 by the Wakamatsu Harbor Improvement Works, with the result that the depth of water alongside the quay wall for about 630 feet is twenty feet at ebb tide. A branch line of the Kyushu Railway has a station immediately adjoining the steel works, and short lines connect the various parts of the works and the quay (amounting in all to thirty miles of lines) with the Kyushu Railway. Coal can be carried by the branch line from the mines at Futase, in the Province of Chikuzen, which belong to the steel works. The present output from these mines is not sufficient to meet the demands of the works, supplementary supplies being obtained from private-owned collieries at Miike, in Chikuzen Province and the Island of Takashima, near Nagasaki. The total consumption of coal per annum is about 500,000 tons. A very complete coal-washing plant is in operation to supply the coke ovens. The cost of coke delivered at the blast furnaces is about twenty shillings per ton. A by-product plant is being built to take ammonium sulphate and tar from the Semet-Solvay Battery.

There are two blast furnaces in working order, and one in course of construction, which will be completed this year. The two furnaces now in use produce 300 tons of pig iron in twenty-four hours, one giving 175 tons and the other 125 tons. At present the blast machinery is driven by steam generated in twenty-four Lancashire boilers by the gas from the furnace tops, but a by-product plant is in course of construction for purifying the gas from the furnace tops, and using it in engines to drive the blast machinery.

The ore used in the furnaces is hematite, with some magnetite and limonite. About eighty per cent. of this ore comes from the Dayeh mines, near Hankow, in China, under special contract with the Hang Yang Iron Works, owners of the mines. The ore contains on an average sixty per cent. of iron, and at least 100,000 tons must be supplied annually. The cost of this

ore at the mines is from 4s. to 7s. per ton, and the freight to Wakamatsu is 7s. per ton, so that the cost of the ore delivered at the furnaces is from 11s. to 14s. per ton. An irregular supply of hematite ore is obtained from Korea, and considerable quantities from different parts of Japan; but the point of importance to be noted is that for purposes of manufacture the coal and iron fields of China are more convenient to the works than those of Japan, and this opens up great possibilities for the future.

There are three principal departments in the works, viz: (1) the pig-iron department; (2) the steel department; and (3) the rolling-mill department. Besides these there are the electric central building, central pumping station, iron foundry, repairing shop, pattern shop and foundry sand storage, boiler shop, smithy, chemical and mechanical laboratory and inspection bureau, and fire-brick plant. The buildings are lighted throughout by electric light.

There are at present two Bessemer converters, with a capacity of 150 tons each per twenty-four hours, one charge amounting to 10 tons. In three years' time a third plant will be completed, according to designs drawn up by the German expert in charge of the Bessemer department. There are now eight Siemens-Martin furnaces, with a capacity of about the same quantity of molten steel per twenty-four hours as the two Bessemer converters, *i. e.*, 300 tons. A great part of this steel is taken in 5-ton ingots direct to the plate mill. Under present conditions the works are able to turn out about 90,000 tons of finished material a year. The original plans were for an annual output of 60,000 tons, which would have satisfied one-half of the demands of that time, but the success of the venture, and the steady increase of Government requirements, have brought about a sensible extension of the original programme. In the course of the next five or six years it is confidently expected that the annual output will amount to 180,000 tons, *i. e.*, double the present output. By far the greatest portion of the products goes to the Imperial Navy Department, the remainder being purchased by the War and Railway Departments. Materials used at the various arsenals in Japan—Tokio, Kure, Sasebo,

Osaka, Yokiska and Maizuru—are largely supplied by the Imperial Steel Works. Practically all the materials for the the building of ships of war are now turned out at the works; it should be noted, however, that armor plate is not made there.

For the Railway Department 40,000 tons of rails are turned out yearly. These rails vary in weight from 9 pounds to 60 pounds per yard, the output of heavy rails being ten times that of light rails. Heavy rails are exported in large quantities to Korea for use on the Korean and connecting systems. The selling price of heavy rails is about £7, 10s. per ton. The prices paid by the three Government Departments for goods purchased from the Imperial Steel Works are not dependent on the current foreign market prices, but are arranged in advance on the basis of the average prices ruling abroad during the immediately preceding five years, due consideration being also given to existing conditions at the works.

The report to which we have referred gives a table showing the rolled products of the Imperial Steel Works. These include all the ordinary forms in use, and the results of the tensile tests show that the quality of the material is very good.

A large proportion of the machinery employed is of German make. In the first instance all machinery was procured from Germany with the exception of some electric cranes of American make, but now England and America are fairly well represented. The pig-iron and steel plant is practically all from Germany. The number of skilled and unskilled workmen employed at the works is about 7,000, with 3,000 coolies, bringing the total number of employees up to 10,000. The daily wage paid varies from 4½d. to 3s. 10d.—"Engineering."

SCREW PROPELLERS.

Read before the Liverpool Engineering Society by T. Sidney Cockrill, M. I., Mech. E.

The object of this paper is to consider the general features of screw propulsion and propeller design, with which, no doubt, many members of this society are already well acquainted, and to bring to notice a few points bearing on the

subject which have occurred to me from time to time. There is no subject connected with marine engineering on which a greater diversity of opinion exists. Eminent members of the profession have, at times, laid down the most conflicting pronouncements with respect to screw propulsion. So it is hoped that this paper will lead to an interesting discussion, from which the author hopes to gather much useful information. The existence of such difference of opinion on the subject is probably due to the fact that little is really positively known of the mechanics of screw propulsion, or the forces set up by the action of the screw. There are so many disturbing factors, such as progression of the wake, and variation of its speed at different points, augmentation of the ship's resistance caused by the propeller itself, skin friction of the blades, effects due to rotation of the race, effects due to position of the propeller, eddies, cavitation and numerous other influences whose effects cannot be calculated, that it seems almost hopeless to attempt to use first principles in solving the problems involved.

The want of exact scientific knowledge is to be deplored, for when one contemplates the very extensive use of the screw propeller, and how largely the speed and economy of the ship depend on its design and proportions, it is seen that the saving which would follow on more perfect knowledge must be very great.

To meet this want, the governments of this and other countries, and one or two shipbuilding companies, have constructed experimental tanks in which the performances of modern propellers and ships of varying forms can be ascertained; and the results arrived at have been found so closely to agree with the actual performances of the full-size propellers and ships, the dimensions and forms of which have been determined by the model experiments, that there is no doubt but that in the future tank experiments will become the practice for determining the most suitable dimensions of propellers. At the present time, when the turbine, with its increased speed of rotation, is replacing to such an extent the

reciprocating engine for vessels of over 15 or 16 knots, modifications in propeller design render previous practice almost worthless as a basis for designing new propellers, and the experimental tank is practically the only means of securing the highest possible efficiency.

The greater part of our knowledge of screw propellers is due to the two Froudes, father and son, who carried out a series of experiments in the Admiralty tanks, extending over some 30 to 40 years.

The principle on which the screw (as well as all other marine propellers) acts, is that the reaction caused by the projection of a stream of water by the propeller in one direction produces motion of the vessel in the opposite direction. And since rate of change of momentum is proportional to the force which causes it, the force exerted by the propeller is proportional to the momentum per second of the stream of water which it projects. Action and reaction are equal and opposite, and therefore the force exerted by the propeller in driving the water astern, is equal to the thrust in the ahead direction; and this again is equal to the resistance of the ship. Using standard units of measurements, this means, in short, that the momentum of the stream projected astern by the screw is the measure of the resistance of the ship.

The momentum of the stream, or "race," as it is called, is the product of the mass of the water projected in one second and its velocity relatively to the surrounding water, or $M = m(v - V)$. And the mass of the race projected in one second in turn depends on the area of its cross section and its velocity relatively to the ship $m = \frac{W}{g} Av$. The momentum of the race is, therefore, equal to $\frac{W}{g} Av(v - V)$ and this is the measure of the thrust of the screw, and also of the resistance of the ship.

The energy expended in setting in motion the water forming the race is lost so far as propelling the vessel is concerned.

This energy is $= \frac{W}{2g} Av\ (v-V)^2$. That is to say, it varies directly as the mass of the water acted upon, but also as the square of its final velocity. If the mass of the water acted upon were doubled, the loss of energy would be doubled; but if the final velocity were doubled, the loss of energy would be increased fourfold. The most efficient propeller is, therefore, that which projects the greatest mass of water astern at the lowest speed.

Theoretically, the lower the sternward speed of the race relatively to the ship v, for a given value of V, the greater is the efficiency; and it follows that the larger A is made, the more efficient would be the performance.

As the sectional area of the steam of water A which is projected astern by the screw type of propeller is, owing to practical circumstances, larger than that projected by any other type, the velocity of the stream v may be less with the screw than with other types; and this is really the fundamental reason why the screw is the most efficient form of propeller. There is, of course, practically only one other type of marine propeller now used—the paddle wheel; but, in spite of certain disadvantages, the screw has proved to be the better type, taking everything into consideration, and is used very much the more extensively.

The formula for the exact value of the thrust of a given screw propeller, even when it is working under the most favorable conditions, has yet to be invented. The leading principles involved in screw propulsion may, however, be defined.

The resistance of the ship, after its inertia has been overcome, is due to the resistance of the water and the resistance of the air. The resistance of the water may be sub-divided into frictional, wave-making, stream-line and eddy-making resistances.

The resistance in a sea-way is further augmented by the wind and waves, and the pitching of the ship. The resistance in smooth water can be estimated with a very small degree of error, but that in a sea-way is beyond the range of mathe-

matics. It may, however, be remarked that length, size and weight in a ship seem to counteract the adverse effects of wind and waves in a sea-way; and we in Liverpool can bear witness to this, having become quite accustomed to the regularity with which our large ocean liners arrive in port under all conditions of weather.

Referring to the disturbing influences acting between the thrust of the screw and the resistance of the ship, it is proposed at this point to consider two of the most important, one of which, it will be seen, is due to the effect of the presence of the ship on the propeller, and the other to the effect of the propeller on the ship. The first is the wake, or the forward motion imparted to the water through which the vessel moves; and the second is the augmentation of the ship's resistance caused by the sucking action of the propeller.

When a vessel moves through the water it tends to drag with it the water immediately surrounding it. This is caused chiefly by the friction between the skin of the ship and the particles of water in contact with it, but also by the stream-line motion and the waves set up by the motion of the vessel. The water which thus tends to accompany the ship is technically known as the wake. The speed of the wake is greater in the immediate vicinity of the sides and bottom of the ship, and becomes gradually less the further it is removed. It is also greater at the after part of the vessel, on account of the motion being gradually impressed on the water as the vessel moves through it; and for the same reason the speed of the wake is greater in the case of long vessels.

The water in which the propeller works is, therefore, not at rest, and the speed of the propeller through the water is not the same as the speed of the ship. If we denote the speed of the ship by V, the speed of the wake by w, and let V_1 equal the speed of the propeller through the water in which it works, then $V_1 = V - w$. That is to say, the propeller advances through the water at a speed *less* than that of the ship by an amount equal to the speed of the wake. The effect of this is that the thrust of the propeller is *greater* than it would be

were it working in still water. The increase in thrust due to the wake means that a portion of that part of the energy developed by the engines which is expended in giving motion to the wake, is returned as useful work in the form of an augmentation to the natural thrust of the propeller.

In twin-screw vessels, owing to the propellers being further removed from the hull, the increase in thrust derived from the wake is less than that in the case of single-screw ships.

The second disturbing influence is the effect the propeller has on the ship behind which it works. When a propeller is revolving and throwing a stream of water in a sternward direction, it has the effect of reducing the pressure of the water in front of it. Owing to this, the pressure of water on the after part of the ship is less than it would be if the propeller were at rest. There is therefore a reduction in the water pressure behind the ship, and the effect of this on her performance is precisely equivalent to an increase of pressure in front, viz.: an augmentation of the resistance against which the ship moves. The propeller has therefore not only to develop an amount of thrust to overcome the natural resistance of the ship, or, in other words, to exert a force equal in amount to that which would be required to *tow* the vessel at the same speed, but it has to exert an additional amount of thrust to overcome that augmentation in the ship's resistance which is due to the action of the propeller itself.

Here again, as in the case of the effect of the wake, there is a difference between twin-screw vessels and single-screw vessels. Twin screws, being further removed from the hull, do not cause so much loss in pressure on the after part of the ship as single screws, and therefore the augmentation of resistance is less.

Coarse-pitch propellers produce a greater augmentation of resistance than fine-pitch propellers; this, perhaps, accounts for the higher efficiency of fine-pitch propellers.

Considering the net result of the gain in thrust resulting from the wake conjointly with the loss due to the propeller augmenting the ship's resistance, it was found by the Froudes

that for most types of vessels the gain due to the wake and the loss due to the augmentation of resistance so nearly balance one another that they may be neglected in most calculations affecting propeller design. The cancellation of these two factors (each of which, taken by itself, would be sufficient to upset any prediction as to performance) is due to the fact that for most types of ships, the conditions which are favorable to causing an increase in gain of thrust from the wake, are also those which tend to an increase in the augmentation of resistance; and *vice versâ*. It must not be forgotten, however, that in the case of vessels of unusual proportions of speed, or other conditions affecting the design of the propeller, it is unsafe to take it for granted that these influences are in equilibrium.

NUMBER OF SCREWS.

In order to obtain the required thrust to overcome the total resistance, it is necessary to have a certain propeller-disc area if the slip of the propeller is to be kept within reasonable limits. The disc area of a propeller is the area of a circle of the same diameter as the diameter of the propeller over the blade tips. To obtain this disc area it may be necessary to have only one propeller; or two, or even three or four propellers may be required. It is the disc area on which the area of section of the stream thrown sternwards depends, the stream being, in fact, an annular column equal in area to the disc area less the area of the boss. In the case of cargo steamers of moderate speed and ordinary draught of water, the thrust required is easily obtained from one screw. The draught of water in these vessels not being restricted, a propeller of large diameter may be used, giving the required area of section of stream without resorting to duplication of screws.

With higher speeds the resistance increases quickly, being as the square of the speed for moderate speeds; but the resistance often varies at a very much higher power of the speed at higher rates of speed; so that in the case of passenger vessels it is necessary, as a rule, to employ twin screws to obtain the necessary thrust.

When the draught of water is limited again, twin screws may be necessary, on account of the draught being insufficient to allow of a single screw of sufficient disc area and adequate immersion.

In the case of very large powers it is deemed inadvisable to transmit the whole power through one line of shafting, the diameter of shaft required being too large to ensure sound forging. And, again, twin screws with two sets of engines obviate to some extent the risk of total disablement at sea.

In the British Navy twin screws are the rule, chiefly on account of there being less chance of total disablement in action, but also on account of greater maneuvering ability (which is, of course, a very important consideration in war-ships), and on account of reduction in the height of the engines, whereby the cylinder tops are well below the armored or protected deck, so minimizing the risk of injury from shot and shell. In the American and German Navies many of the battleships have triple screws driven by three sets of engines, the center engine and propeller alone being used when steaming at cruising speeds.

The practice of fitting two or three propellers on one shaft for high-speed vessels has, it seems, been abandoned, probably on account of the interference which must inevitably take place. It is noteworthy that in the case of some high-speed vessels (the *Turbinia*, for instance), which had originally two or three propellers on each shaft, a single propeller of larger diameter has been fitted to each shaft, and better results have been obtained.

NUMBER OF BLADES.

There is not much doubt but that a propeller with two blades is more efficient under favorable conditions than propellers with three or four blades, but its efficiency is greatly impaired in a sea-way. The higher efficiency of the two-bladed propeller is due to absence of interference between the blades, and to there being less edgeway resistance. Experience has taught us, however, that on the whole, in actual practice three blades are best for propellers which can be well

immersed below the surface of the water, such as the relatively smaller screws of vessels with two or more screws; and that four blades are best for the large propellers necessary for single-screw ships, these being comparatively near the surface of the water.

SHAPE OF BLADE.

In designing the shape of the blade, it should be borne in mind that whereas blades having their greatest breadth comparatively near the tip, develop more thrust, yet this advantage is practically nullified by the greater frictional resistance of blades of this shape, owing to the broadest part of the blade being situated near the part of greatest circumferential velocity.

The number of patents taken out for shapes of propeller blades alone is legion, but experience seems to prove that the elliptical shape is preferable for most ordinary cases. The Admiralty have long adopted this shape for all classes of warships, and there seems to be no reason in the majority of cases for departing from it.

Speaking generally, the shape of the blade has little influence on propulsive efficiency, but vibration may be reduced by having a suitable shape. The chief considerations as regards efficiency are the relations between diameter, pitch and surface. It is more than probable that the shape of the section of the blade is of greater importance than the shape of the blade itself. The blade should be as thin as possible consistent with strength, so as to reduce its edgeway resistance as it revolves in the water. This rule applies with increasing force from the root towards the tip, the circumferential velocity being higher as the tip is approached, and the resistance increasing proportionately as the square and still higher powers of velocity.

The edges of the blade should be fine and sharp, and the shape of section of the blade should conform to the natural stream lines for the speed intended, so as to prevent eddies and reduce edgeway resistance to a minimum.

The blades of high-speed vessels are usually made of bronze,

owing to the strength of this material allowing of very thin blades, and to its being non-corrosive; the blades have knife-edges, and are polished on both sides.

The use of bronze for propeller blades for all services should, the author thinks, have more consideration than it appears to have at present. Bronze blades may be made much thinner than cast-iron blades, they have smoother surfaces, the edges preserve their keenness, and they do not deteriorate by corrosion and pitting, so that the resistance of the propeller itself is very much reduced; and, after all, there is the intrinsic value of the bronze as scrap to be taken into account when considering the increased first cost.

DIAMETER OF PROPELLER.

The diameter of the propeller is fixed by the required disc area, about which something has already been said when considering the number of screws. Theoretically, the efficiency rises with increase in the weight of water acted on, so that the larger the diameter the higher will be the efficiency. The diameter should never be so large, however, as to cause the tips of the blades to break through the surface of the water, or even to be near the surface under ordinary trim, as then the blade carries air down with it, and the presence of air in the race is the cause of a great loss in efficiency. The mixture of water and air is not only of lower density, but the presence of air lowers the speed at which water will flow to the propeller, and the result is racing or acceleration of the speed of rotation. Racing in a sea-way is, the author believes, as much due to the presence of air in the race as to the blades being partly out of the water.

A circumstance which tends to upset the theoretical rule that the larger the propeller the higher will be the efficiency, is that the resistance of the blades increases more and more rapidly with each unit of increase in diameter. A large-diameter propeller, while being efficient owing to the large amount of water acted upon, may absorb so much power by its own resistance that it would not give such good results as

a propeller of smaller diameter. The point is, in fixing the diameter, to ascertain at what point in the scale of diameters the rapidly-rising curve of resistance overtakes the less rapidly rising curve of theoretical efficiency. The efficiency of reciprocating engines, however, usually rises with decreased revolutions, so that even the "ultimate" efficiency of the propeller may be sacrificed to some extent in order to gain in the total efficiency of the engine and propeller combined.

Too small a diameter results in an abnormal amount of slip and consequent waste of power.

Vessels with bluff lines require larger-diameter propellers than vessels with fine lines, the water in which the propeller works being more disturbed. With a propeller of large diameter the blades reach out into more solid water.

Referring to propellers driven by turbine engines, the thrust developed is proportional to the square of the diameter, D^2, and also to the square of the velocity of the stream driven sternward, presumably $= (P \times R)^2$. These terms are reciprocals of one another to obtain the same thrust. But if the diameter be made too small, and the $P \times R$ too great, a loss in efficiency will result, owing to an abnormal slip. In the engines the velocity of the turbine blades is fixed almost beyond consideration, being about 0.5 that of the steam flowing through the turbine. The mean velocity of the blades $V = D_B \pi R$; where $D_B =$ the diameter of the rotor taken half way along the blades. It will be seen that an increase in revolutions R means a decrease in the diameter of the rotor D_B, and consequent saving in the weight and space occupied by the turbine. It is therefore very desirable to run the turbine at as high a speed of rotation as possible, but as this also means reduced diameter of propeller and consequent loss in propeller efficiency, a compromise must be made so that the combined engine and propeller efficiency may be as high as possible. It appears probable that a higher combined efficiency would result if slightly larger turbines than those in use at present, with corresponding decrease in revolutions and increase in propeller diameter, were adopted.

The point to be inculcated with regard to propeller efficiency is the absolute necessity of a free and unrestricted flow of water to the propeller; and this can only be obtained by molding the run of the vessel as fine as possible, and of such lines as not to interfere with the natural stream-like motion of the water as it closes in under the stern.

It may be remarked here that all obstructions to the free flow of water round the propeller boss should be avoided, especially for high speeds. The nuts for securing loose blades should be in recesses, with plates fitted over, to preserve the contour of the boss. A conical cap should always be fitted at the after end of the boss. The length and shape of this cap should be designed to suit the stream-lines abaft the boss for the intended speed, for if the cap does not *entirely* fill the cavity formed by the water behind the boss it will not serve its purpose, and may just as well not be fitted. Means should also be provided for preventing eddies at the after ends of the stern tubes and other places, and for preventing the too sudden parting of the water at the fore end of the propeller boss.

BLADE AREA.

The area of the blades, or blade surface, is conveniently expressed as a fraction of the disc area.

The following table gives the ratio of blade area to disc area which is generally adopted:

Cargo vessels with full run, single screw,	.31 to .34
Cargo vessels with medium run, single screw,	.34 to .37
Cargo vessels with fine run, single screw,	.37 to .39
Launches, single screw,	.34 to .36
Fast passenger vessels, twin screws,	.40 to .43

It is generally thought inadvisable to go beyond .5 or .6 on account of causing interference between the blades; but in some high-speed turbine vessels the ratio is considerably higher than this.

PITCH.

The pitch is sometimes made to increase gradually from the root to the tip of the blade; and in some propellers the pitch is uniform over the blade except near the boss, where it is finer. The object of a decreased pitch near the boss is to minimize the churning of the water which is supposed to take place at that part.

Some makers, on the other hand, cause the pitch to increase *across* the blade—finer for the leading or forward half of the blade, and coarser for the following or after half. The idea in this is that the water driven astern is gradually accelerated. This can only be recommended in the case of very wide blades with high pitch ratio.

Experience shows more and more, however, that little advantage, if any, is gained by a variable pitch; and uniform pitch is becoming the standard practice with most makers. The Admiralty seldom adopt any kind of variable pitch.

SLIP.

The slip is the difference between the speed of advance of the propeller (supposing it to be working in an unyielding substance) and the actual speed of the ship. In other words, it is equal to the pitch multiplied by the revolutions, less the distance traversed by the ship. If the water did not yield to the propeller and flow sternward, the speed of the ship would be the same as the speed of the propeller, and there would be no such thing as slip; but water, being a fluid, is driven astern by the action of the propeller as the ship moves ahead. The rate at which the water is driven astern relatively to the surrounding water is usually said to be equal to the slip; but this is only true provided the pitch multiplied by the revolutions is equal to the speed of the race relatively to the ship, or, in other words, provided the propeller itself does not slip in the race. As far as the author can see, we have no means of ascertaining the truth of this.

The above, however, only relates to *apparent* slip, for it does not take account of the fact that the propeller is not

working in still water, but in water in motion in a forward direction owing to the influence of the ship in passing through it; and, as it is the *propeller* that is under consideration, the speed of the propeller and not the speed of the ship through the water should be the basis in calculating the slip.

For example, if the pitch were 15 feet, the revolutions 81 per minute, and the speed of ship 11 knots: Then

Speed of propeller $= \frac{P \times R \times 60}{6,080} = \frac{15 \times 81 \times 60}{6,080}$ $= 12$ knots

Speed of ship, ... $= 11$ knots

Apparent slip of propeller, ... $= 1$ knots

But supposing the wake to have a speed of 2 knots: Then

Speed of propeller as before $= \frac{P \times R \times 60}{6,080}$ . . $= 12$ knots

Advance of propeller through the water in which it works $= V - w = 11 - 2$, ... $= 9$ knots

"Real slip" of propeller, ... $= 3$ knots

It is sometimes found that a propeller seems to have *negative* slip, that is to say, that the speed of the ship is (apparently) greater than the speed of the propeller which drives it. This negative slip is only found when the speed of the ship is taken instead of the speed of the propeller through the water; it is the apparent slip, not the real slip, of the propeller.

The phenomenon of negative slip is the cause of a vast amount of ingenious, if not always scientific, hypotheses to account for its existence. Everyone agrees that negative real slip is a physical impossibility. In the author's humble opinion negative apparent slip is also a physical impossibility, and therefore there is no need to account for a thing which does not exist. This is on the understanding that "apparent slip" means $\frac{P \times R}{60} - V$ while "real slip" means $\frac{P \times R}{60} - (V - w)$; these are the usual definitions of the terms, and they are given here to prevent misunderstanding arising from the mere

meaning of words. The explanation is simple enough, for in order to calculate the slip, the pitch P must first be known; it is one of the factors in the calculation, and, in the author's opinion, no one knows what the true effective pitch of a propeller is. What is usually taken as the pitch of a propeller is the pitch of the after face of the blade, or, rather, the mean of the various pitches found at different parts of the after faces of the blades. This is certainly not the effective pitch of the propeller with respect to its action on the water, one reason at least for this being that it takes no account of the curved back of the blade, and it is certain that this increases the true effective pitch, although no one knows exactly, or even approximately, what value is to be assigned to it. There are also other reasons for supposing that we do not know what the effective pitch of a propeller is.

The cause of negative slip often given is that the propeller is working in the wake of the ship, and therefore working in water which has a forward motion.

For example of this reasoning we may take a previous example. Supposing the speed of the ship to be 13 knots and the speed of the wake 4 knots.

Then speed of propeller, as before, . . .	12 knots
Speed of ship, ...	13 knots
Apparent slip of propeller, ...	—1 knot
And speed of propeller, as before, . . .	12 knots
Advance of propeller through the water = 13—4 =	9 knots
"Real slip" of propeller, ...	3 knots

This reasoning seems to take it that the forward motion of the wake is greater than the backward motion imparted to the water by the propeller, so that the water forming the race still has a slight forward motion after it has been acted upon by the propeller. We would then have a large volume of water (the wake) following the ship, *part* of that volume (the race) is *impeded* only (not even brought to rest) by the pro-

peller. The net result of the wake and the race would be motion in the *same* direction as the motion of the ship. But the motion of the ship is due to re-action on the water, and therefore the net result must be water flowing sternward. The author thinks there is no doubt whatever about this, and that any theory which supposes that the net result on the water of a self-propelled vessel is motion of water in the same direction as the motion of the vessel, is condemned from the outset.

An explanation of negative slip put forward by Mr. Barnaby is based on the fact that the feed water to the propeller is accelerated while still forward of the propeller; and the action of the propeller upon the water while in contact is to accumulate pressure, which has the effect of increasing the acceleration of the race after it has left the propeller; and that, owing to this, the slip or acceleration of the water in the race is always in excess of the slip of the screw as given by its "nominal" pitch. But it rather seems to the author that this not only explains negative slip, but it explains it away, for it means that what we call the pitch of the propeller is not the true effective pitch.

In connection with slip there is a paradox which should be noticed. It is quite true that the most efficient propeller is that which projects the greatest mass of water astern at the lowest speed. This is equivalent to saying that the most efficient propeller is that which has the least slip. But we know by experience that a very small amount of slip, or no apparent slip, or negative apparent slip, is a sure sign of inefficiency; and that a propeller giving no slip, or negative slip, should be changed as soon as possible, as it is not suited to the vessel. How are these apparently contradictory statements to be reconciled? Both are true; but while in the first statement efficiency is taken in a general theoretical sense, neglecting friction, or, in such a sense, for instance, as we should use if we were comparing the theoretical efficiency of screws and paddle wheels, or screw propulsion and jet propulsion, or even screw propulsion itself applied to different vessels, in the second

statement efficiency is taken with respect to the actual results (which include friction and a host of other unknown and incalculable factors) of a particular case. The explanation stated less generally is this: Theoretically, a propeller having a small amount of slip is efficient; but smallness of slip is invariably due to fineness of pitch; and the finer the pitch the greater the loss by surface friction, until a point is reached at which the loss by friction overcomes the gain in the theoretical efficiency, and any further reduction of pitch produces a loss in what may be called the actual efficiency.

CAVITATION.

In very high-speed vessels such as destroyers, the troublesome phenomenon termed "cavitation" is likely to occur if the blade surface be too small for the thrust developed. Cavitation is failure in the supply of water to the propeller, and results in the formation of a vacuum or cavity at the forward faces of the blades owing to the water being unable to follow the blades fast enough. It was first discovered by Mr. Barnaby during the trials of H. M. destroyer *Daring* in 1894. It may be likened to what would occur in the case of a pump plunger which rises so quickly that it exerts a pull on the water beneath it of more than 15 pounds per square inch. In that case the pressure of the atmosphere is insufficient to cause the water to follow the plunger, and a vacuum or cavity is formed. In the case of propellers a cavity is formed on the forward side of the blade if the peripheral speed is too high, on account of the head of water being insufficient to cause it to follow the blade; and the consequence is a reduced thrust.

At the surface of the water, cavitation would take place when the pull of the blade on the water exceeds the pressure of the atmosphere; and below the surface, when the pull exceeds the pressure of the atmosphere plus the pressure due to the head of water. This, however, leaves out of consideration, for the sake of simplicity, the fact that, owing to the water giving off vapor when the pressure is reduced, cavitation actually takes place when the pull is somewhat less than this.

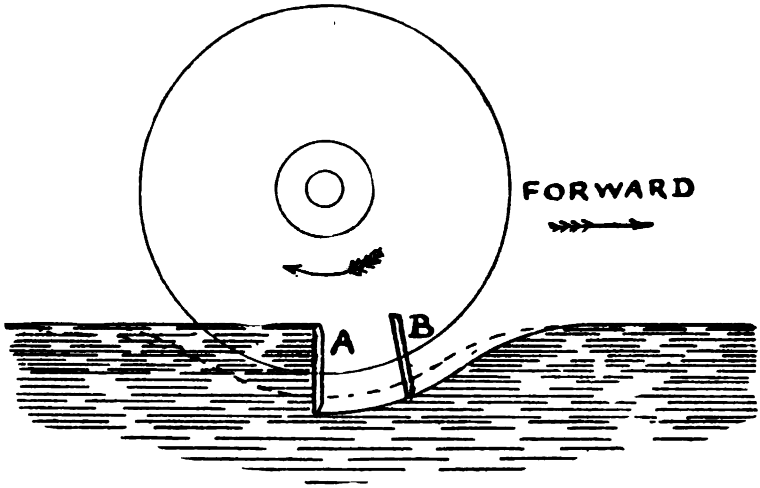
It commences to take place at the tips of the blades (where the circumferential velocity is highest) as they pass the highest point of their path where the pressure of water is least.

In destroyers and other high-speed vessels the projected blade area should not be less than that which will insure the thrust per square inch being less than about 12 pounds, otherwise cavitation may ensue.

Cavitation may be detected by a sharp ascent in the slip curve above a certain speed.

The thrust exerted by a propeller depends on the difference in pressure on the after and forward sides of the blades. It is therefore desirable to have as great a pressure on the after side and as small a pressure on the forward side as possible. Reduction of pressure on the forward side of the blade is equivalent to increase of thrust, and is to be sought for. It would therefore seem, on the face of it, that if the propeller revolved so rapidly that the water was unable to follow the blades, so causing the pressure on the forward side to fall to zero, a highly satisfactory state of things would be arrived at. But further consideration will show that if the water does not follow the blades the pressure on the after side will also fall, as there would be a lack of water there also, so causing a greatly diminished thrust. It is a *sine qua non* that to obtain efficiency the water must be in contact with the forward side of the blade.

Cavitation is more easily grasped by considering the paddle-wheel. Referring to the sketch, which shows a hypothetical case for the sake of simplicity we will suppose we are dealing with only two floats A and B. The float A enters the water so rapidly that a cavity is formed forward of it. So far as this float is concerned, a highly satisfactory state of things is arrived at; for we have a great pressure of water on the after face of the float and no pressure on the forward face, tending to create a great amount of thrust to drive the vessel ahead. But it is apparent that the float B has no water to engage with, and a loss in thrust is therefore inevitable. (What would actually happen in a real case, with the full number of



floats and the vessel moving ahead, is that the water line in the vicinity of the wheel would probably take the form shown by the dotted line, that is to say, all the floats would be only partially but equally immersed.)

Almost the same sort of thing happens to the screw propeller; you cannot have great pressure on the aftersides of the blades while there is a cavity on the forward sides, owing to an insufficient quantity of water flowing past them.

Cavitation is due to the slip of the propeller; if there were no slip there could be no cavitation. This will be more easily seen by referring to the sketch of the paddle wheel. In this paper the author has touched upon a few of the leading features only of screw propellers and propulsion; to deal with the subject at all fully would be impossible in a short paper. It is the most recondite and difficult subject that marine engineers have to deal with with, and it gives rise to the greatest difference of opinion. But the author feels sure that, in time, the apparent vagaries of the screw will be entirely mastered by theory and reduced to mathematical formulae; and then we may expect marine engineering to progress by leaps and bounds.

SHIPS.

UNITED STATES.

· **Viper**—*Launch of.*—The submarine *Viper* (*No. 10*) was successfully launched March 30, 1907, at the works of the Fore River Shipbuilding Co., Quincy, Mass.

Tarantula—*Launch of.*—The Submarine boat *No. 12*, the *Tarantula,* was launched at the works of the Fore River Shipbuilding Co., March 30, 1907. This is the last of the four submarines building by the Electric Boat Company to be launched.

Birmingham.—The Scout Cruiser *Birmingham* was successfully launched at the works of the Fore River Shipbuilding Co., Quincy, Mass., on May 29.

The *Birmingham* is one of the three scouts authorized by Act of Congress, approved April 27, 1904. The other two are the *Salem* and the *Chester.* The *Salem* is also being built by the Fore River Shipbuilding Co., while the *Chester* is building at the Bath Iron Works, Bath, Me.

The following are the chief characteristics of the *Birmingham.* Displacement, 3,750 tons; length, 423 feet; beam, 46.6 feet; draught, 16¾ feet; I.H.P., 16,000; armament, twelve 3-inch guns; torpedo tubes, 2; speed, 24 knots; coal, 1,250 tons.

The propelling engines will be right and left, placed in watertight compartments and separated by an athwart-ship bulkhead. They will be of the vertical, inverted-cylinder, direct-acting, triple-expansion type. The general particulars of engines will be as follows:

Diameter of high-pressure cylinder, inches	28½
intermediate-pressure cylinder, inches	45
forward low-pressure cylinder, inches	62
after low-pressure cylinder, inches	62
Stroke of all pistons, inches	36
Indicated horsepower of both main engines	16,000
Revolutions	200
Steam pressure, at high-pressure receiver, pounds	250

The main valves will be worked by Stephenson link motions with double-bar links. There will be one piston valve for the high-pressure cylinder and two each for the intermediate and low-pressure cylinders.

The framing of the engines will consist of forged-steel columns trussed by forged-steel stays. The engine bed plates will be of cast steel, supported on the keelson plates. All crank, thrust, line and propeller shafting will be hollow. The shafts, piston rods, connecting rods and working parts generally will be forged steel as specified.

Each main condenser will have a cooling surface of about 8,000 square feet, measured on the outside of the tubes, the water passing through the tubes.

The main circulating pumps will be of the centrifugal type, one for each main condenser. For each main condenser there will be an independent air pump of the double, vertical, double-acting type.

There will be an auxiliary condenser, of about 600 square feet cooling surface, in the forward engine room, connecting with all the auxiliary machinery.

There will be an auxiliary condenser of about 200 square feet cooling surface in the dynamo room for the exclusive use of the dynamo engines.

The auxiliary and dynamo condensers will have air and circulating pumps.

There will be twelve water-tube boilers of the "Express" type, placed in three watertight compartments, as shown.

The total grate surface will be at least 693 square feet.

The total heating surface will be at least 37,080 square feet.

The forced-draft system will consist of blowers discharging into air-tight firerooms, there being two blowers for each boiler compartment, as shown.

There will be four smoke pipes, as shown.

There will be a main feed pump, a fire and bilge pump, and a bilge pump driven from the forward end of the crank shaft in each engine room.

There will be a feed pump and a fire and bilge pump in each boiler room.

The starboard propeller will be right-hand and the port propeller left-hand, the engines turning outward when going ahead.

There will be ash hoists, ash ejectors, turning engines, auxiliary pumps, refrigerating machinery and motors or steam engines, workshop machinery and motors, distilling and evaporating apparatus, steam reversing gear with oil-control cylinder, and such other machinery, tools, instruments or apparatus as are hereinafter described, together with such other as may be necessary to complete the machinery in accordance with the specifications and plans as finally adopted.

All parts of machinery and boilers will be secured in an approved manner to prevent displacement when the vessel is used for ramming.

Pipes, gear and other fittings will be so arranged that the tubes of boilers, condensers, evaporators, feed heaters, distillers, etc., may be withdrawn without interference, and so that parts of main and auxiliary machinery may be easily removed for examination and repair.

See JOURNAL A. S. N. E., Vol. XVII, p. 263, for a full description of this vessel.

Submarine Tests.—*Octopus and Lake.*—Interesting trials between the submarine boats *Lake,* built by the Lake Torpedo-boat Co., of Bridgeport, Conn., and the *Octopus,* representing the Holland type of boat, built by the Electric Boat Company, of New York, began at Newport, R. I., May 1. The *Octopus* was the first boat to start on a test on May 1. She had to abandon the trial, however, because she broke a bracket of her port engine just as she was about to be sent over the measured-mile course. Before the accident the *Octopus* had several successful unofficial runs over the course. The members of the naval trial board were aboard the *Octopus* when the bracket broke.

The *Lake* underwent her first official test on May 2, which she completed without any mishap, in the most successful manner, with the members of the trial board aboard. The conditions of this day's test were that the boat should make three

surface runs under both her motors and her gasoline engines and three under her gasoline engines alone. A strong northeast wind was blowing down the course and, with the tide setting in, the sea was as choppy as it ever is on the bay except in storms.

Her unofficial time is given as eight and six-tenths knots an hour. Under her gasoline engines alone the reduction was hardly one knot, the rate being 7.68 knots. It is said that tidal corrections will give the *Lake* an even better record, for the current which she faced for some time was strong and for a time she had to encounter a choppy sea.

The *Lake* reached the upper line of the course in 7 minutes and 16 seconds, a rate of 8.26 knots. Running back, the boat had the advantage of the wind. but the tide was against her and the sea rough. The *Lake,* however, made a fine run, her time being 6 minutes and 33 seconds, or 9.16 knots per hour. The average of these two runs was a rate of 8.71 knots. The run up the course again, which was against the wind, but with a little tide favoring the boat, was made in 7 minutes and 43 seconds, a rate of 7.83 knots. Down and up the course the average speed was 8.49 knots. This completed the try out of the boat at a maximum speed under both her motors and her gasoline engines, and it was found that the average rate was 8.6 knots an hour.

In the speed tests of the *Lake* under gasoline engines only the conditions of wind and sea had improved, and on the third run down the bay, with wind and tide with her, she made her best speed of the day, which was at the rate of 9.18 knots. Her first run under engines was made in 6 minutes and 57 seconds, a rate of 8.63 knots. Her second run, against both wind and tide, was her slowest, the time being 9 minutes and 18 seconds, and the rate 6.45 knots. The average rate of these two runs down and up the course was 7.54 knots. The mean average speed for the three runs with gasoline engines alone was 7.68 knots. Five more runs under the gasoline engines alone were made at gradually reduced speed to enable the Board to make a speed curve from which

to figure exact deductions. The *Octopus* having been repaired was to undergo her trials on May 2.

Advantage will be taken of our increased knowledge of submarine construction to so alter the *Pike* and *Grampus,* which have been so long at the Mare Island Yard, as to greatly increase their factor of safety. Larger batteries will lengthen their period of possible submersion to three hours, and bilge keels will enable the crew to move about more freely without altering the displacement of the vessel. Hooks for hauling the vessels to the surface in case of accident are to be placed on the side.

The *Octopus* in her first trial on May 3 broke two speed records while going over the Government's measured-mile course. In three times over the course the boat made an average speed of a fraction over eleven knots, while using only her gasoline engines. Besides being tested for surface runs, the *Octopus* had a series of trials in the semi-submerged condition, and in these made ten knots, with electric motors alone in use. To show the *Octopus'* ability to go under water quickly, the boat made a dive, going down twenty feet and immediately coming to the surface again.

When she had finished her first run and her speed had been checked up it was found that she had made the mile in 5 minutes 11 seconds, which was a rate of speed of 11.6 knots. Both down and up the course, or against and with the tide, the *Octopus* had made a rate of 10.78 knots, and her mean for the three runs at maximum speed as required by the specifications was a small fraction over eleven knots. Running semi-submerged, or awash, the *Octopus* gave another fine exhibition, attaining a maximum of 11.5 knots with a strong tide with her, and a mean of 9.98 knots.

More surface runs were made at gradually reduced speed to enable the Board to prepare a speed curve of the *Octopus.* The *Octopus* had a strong tide in her favor in making her first run awash, which was down the course. Only her electric motors were used, and her speed under the conditions was perhaps more surprising than it had been on the surface. The

mile was made in 5 minutes and 26 seconds, which was a rate of speed of 11.05 knots. Up the course the *Octopus* encountered a strong tide, but she was able to make the mile in 6 minutes and 41 seconds, the rate in knots being 8.97. The round trip both with and against the tide was made at a rate of just about 10 knots. Down the course again with the tide the *Octopus* made the run in 5 minutes and 29 seconds, a rate of 10.94 knots.

Her owners had claimed only 10.5 knots for her in cruising trim, and her performance gave a general surprise. Capt. Simon Lake, the inventor and the attachés of the *Lake,* were among the first to give their competitor credit for her record. Captain Lake is, however, greatly pleased with the performance of his own boat on May 2. He has never claimed great speed for her, but is relying on other advantages and facilities which have attracted attention.

Under perfect weather conditions the *Octopus* underwent her second test on May 8, and made what is claimed to be a world's record by attaining a speed of more than 10 knots under water. The best time heretofore made was 8½ knots by one of the submarines of the British Navy. The *Octopus* made three high-speed runs under water with Captain Marix on board. She glided through the water with apparent ease and covered the first mile at the rate of 10.15 knots. This was her maximum speed, while her average was 10 knots plus, which was only a fraction less than she made May 3, while running semi-submerged. According to the conditions the top of her conning tower had to be at least ten feet below the surface, which meant that those aboard would be at least twenty-three feet below. She had, however, to show at least three feet of a mast above the water as a marker for the observers, and from the top of this a little silk flag fluttered. After her submerged run the *Octopus* was put through several tests which came under the head of maneuvering or handling. The conditions require that the time be taken in diving to certain depths and returning to the surface, and in this series the *Octopus* made two porpoise dives. She went down at an eight

degree angle to a depth of 26 feet in a fraction less than 40 seconds. Then she returned to the surface so that an observation of 5 seconds could be made from her conning tower, and made another dive and another broach under the same conditions and in the same time. The best previous record for such diving was 46 seconds, made by the submarine boat *Fulton* of the *Octopus* type.

A feature of the tests was the use for the first time of the submarine bell, with which the *Octopus* is equipped. Another bell and receiving instrument were attached to the tender *Starling*, and communication was kept up all the time the submarine was under water.

The *Lake*, which was to have begun her submerged trials on May 8, sprung a leak in her torpedo tubes, and it was decided to postpone her trial until the damage was repaired.

In the trial of the submarines in Narragansett Bay, R. I., May 10, before the Navy Trial Board, the submarine boat *Octopus* established another record in making quick submergences while going through maneuvers in the government tests. The test was made in a gale blowing forty miles an hour. The boat, while going at full speed on the surface, twice shifted her motive power, adjusted her diving rudders and dived to a depth of twenty feet in an average time of 4 minutes and 30 seconds. The best time in which any submarine now in the Navy has ever gone through the same maneuvers is 11 minutes. The boat was also put through a series of turning trials, both on the surface and beneath, to determine her tactical diameter. The board tested the air compressors on the *Octopus*, which have to stand a pressure of 2,000 pounds to the square inch. They stood the test satisfactorily.

The *Octopus* had another test of endurance on May 11 in a ten-mile run while partially trimmed or in diving condition, and an average speed of 9.89 knots was attained on the entire run, which was made with the gasoline engine as a motive power. On this run the engines, according to the figures of the trial board, made forty-two revolutions more a minute than

the standardization figures called for. In trimming for the run the *Octopus* occupied a little over a minute.

Aside from the government test the *Octopus* had two extra tests at the request of the builders. These included six runs over the course, three under one engine, while the other was being used to charge the storage batteries, and the other three while the boat was being propelled by only one propeller. On both of these tests the boat made nearly eight knots.

A successful test was made on May 13 in connection with the trials of a submarine life preserver, cap and jacket. The apparatus resembles an ordinary diving suit, the air being supplied by a tank of oxylite contained on the inside of the jacket. A member of the *Octopus'* crew was submerged in one of the suits for four minutes. The *Octopus* made a run of thirty-five miles by storage-battery power to determine the distance she could go without recharging. According to the figures of the board, the boat could run submerged 115 miles with one charging of the batteries.

On May 14 the *Octopus* was subjected to various maneuvering and special trials and torpedo firing tests. Among the maneuvers were trials of the boat's ability to remain at rest while submerged, turning trials under various conditions and motive powers and quick stops. In the latter the boat came to a full stop from full speed ahead in 50 seconds. Of the special tests the most important was of the automatic devices for blowing out the water ballast to allow the boat to come to the surface in case of accident. In this test the *Octopus* came to the surface from a depth of 40 feet in 43 seconds. Only two torpedoes were fired. These were discharged as the boat was about to dive to a depth of 20 feet. An irregularity with the tubes prevented the discharging of the remaining two after the boat had submerged. Only one of the torpedoes was recovered, the other not coming to the surface.

Both the submarines *Octopus* and *Lake* began their twenty-four-hour submergence test on May 15 at the naval coaling station at Bradford in about 30 feet of water. The *Octopus* carried down a crew of sixteen men, while nine men went down

in the *Lake*. Both boats came up to the surface on the afternoon of May 16 after their twenty-four-hours' continuous submergence test, and none of the men on board of them looked any the worse for being bottled up under the sea for twenty-four hours. Incidentally new records of sustained life under the sea had been made, the seventeen hours for which a party on the submarine boat *Fulton* was submerged at New Suffolk, L. I., in the spring of 1902, having been the previous record. Later in 1902 a party lived twelve hours on the *Fulton* submerged at the torpedo station.

The scientific results of the interesting experiments will not be reduced to figures for several days. A sample of the air in each boat was taken in a bottle at stated periods, and this will be examined for carbonic acid gas and chloride gas, which are very noxious and may have been generated during the four hours the engines were under way.

Captain Cable, of the *Octopus,* said that he had used about one-forty-fourth of the stored air in his boat, from which he estimated that so far as air was concerned he could have remained below for three days longer. He said they ate and slept below just about as they would have done had they been in their homes. Captain Lake, of the submarine *Lake,* said: "We could easily have remained submerged for another twenty-four hours. We used a comparatively small supply of compressed air—in fact, not much more than two hundred pounds, and this was not applied until near the end of the test. If we had used oxone, of which we had plenty on board, and if we had been allowed to open our diving door to admit water to the diving compartment with which to force the carbonic acid gas, we would have drawn very little, if at all, upon our supply of compressed air. With all the *Lake's* various air-disposing arrangements in use a crew in the boat could easily stand a submergence of three days."

The submarine *Lake*, on May 17, went out on a series of maneuvers for three hours to prove the worth of her engines and motors and also her mobility under various conditions. Later in the day she began a thirty-hour endurance run in the

bay, and in all she gave an excellent account of herself. One of the *Lake's* reversing trials attracted much attention from the fact that the reverse was made by merely changing the pitch of her screws, which rendered stopping or reversing her engines unnecessary. Going ahead full speed on the surface under the power of both her engines she was reversed by this changing of pitch and became dead in the water in one minute and three seconds. As a result of her thirty-mile endurance run it was observed that her radius of action under her gasoline engines, carrying the normal amount of fuel, was about 440 miles. She cruised up and down the bay at the rate of about 7.25 knots, at which it took her four hours and thirteen and one-half minutes to cover the distance. Of her normal store of gasoline, which is about 1,500 gallons, she used about 110 gallons. She has emergency tanks, however, which enable her to carry about 4,500 gallons, with which her radius of action would, of course, be greatly increased. Her machinery stood the test of the long journey well.

The *Cuttlefish,* one of four submarines built at the Fore River Shipbuilding Company's yards, at Quincy, Mass., by the Electric Boat Company for the Government, was subjected to a 200-foot submergence test about six miles off Boston Light on Saturday afternoon, May 18, and the boat stood the test without damage or strain of any kind. No member of the crew went down in the boat, it being lowered below the surface by a large derrick vessel. Three submergences were at 150, 175 and 200 feet depths. When it was drawn up a very critical examination was made of it, and it was found that not a drop of water had leaked in anywhere, and that her machinery, which was carefully inspected, was in perfect condition. At a depth of 200 feet the pressure on the boat was about ninety pounds to the square inch, while on the whole boat there was a combined pressure of 15,000 tons. The *Cuttlefish* is the same type of boat as the *Octopus,* now under trials at Newport.

Trials with the *Octopus* were resumed on May 20 in Coddington Cove for torpedo practice. She carried four torpedoes and fired three at a target supposed to be a battleship,

marked by two rowboats 500 feet apart. The range was about 800 yards. The torpedoes were fired while the boat was running in a submerged condition, taking observations by the periscope and going at full speed. The first torpedo, fired from the port tube, ran straight to the target, then turned to a sharp angle to the right. The second got hung up in the starboard tube. The third torpedo went straight, but did not carry more than 500 yards. The *Octopus* also underwent anchor-gear trials and a successful test of her wireless apparatus. Later in the day, when the *Octopus* was at her slip at the torpedo station, she fired another torpedo from her starboard tube along the breakwater, and it went in good shape, apparently showing that there was something wrong with the torpedo previously fired from that tube. One of the torpedoes was lost, making the second lost by the *Octopus* within a week. They are valued at $3,000 each.

The submarine boat *Lake* had her submerged speed trial in Narragansett Bay May 21, and was also put through other maneuvering tests. Previous to the trial, the *Lake* and the *Octopus* were taken out to Brenton's Reef Lightship with the idea of giving them a sea trial in rough water. The wind was strong, but it was offshore, and the sea was comparatively smooth. The boats then returned to the harbor, and the *Lake* was sent over the measured-mile course three times. On the first run, with the tide and against the wind, she covered the course in 10 minutes and 29⅗ seconds, a speed of 7.72 knots an hour. The next run was made against tide and with the wind, and the *Lake* developed a speed of 5.69 knots an hour. On the third run against the tide the speed was 5.61 knots. The *Lake* was then sent over a course on which full speed was given her and then required to stop and back, a feat which she accomplished in 42 seconds. She made the maneuvers of sealing her top and dropping to twenty feet in 7 minutes and 19 seconds, and followed this by anchoring in forty feet of water, as required by the Board. The boat then returned to the harbor.

The *Lake* had her torpedo-firing trials and a run in the open sea on May 22. Her torpedo-launching outfit worked per-

fectly, each of the three projectiles being started off smoothly and, except in one instance where the discharge was evidently premature, the directions were accurate. The firing took place in Coddington Cove, two small boats being placed three hundred feet apart as a target. The target was made by two boats, anchored three hundred yards apart, to represent a battleship. The range was 800 yards. Observation of the target for the first two shots was made by the armiscope, while the *Lake* was submerged, but on the third shot observation was made from the conning tower. The first shot left the starboard tube all right, but the starting lever of the torpedo failed to trip. The second shot, fired from the port tube, was a success, the torpedo crossing the line between the two boats. Then the *Lake* started to turn to fire a torpedo from the stern tube, but the torpedo was discharged prematurely, and made only a short run through the water. All these torpedoes were later picked up, but there was no reloading on board. The *Lake* next executed some maneuvers at a depth of ten feet, completing a circle in 3 minutes and 35 seconds. Late in the afternoon the *Lake* was ordered to sea for a trial in light and awash conditions. The boat went through the test well, although the sea was running high.

The *Lake,* on May 23, gave a wonderful performance of submerging herself, and with her crew on board went five feet deeper than any boat has ever done, with human beings in her, thus establishing a new record. She went down 135 feet. The world's record previous to this was 130 feet, made by a French submarine. The *Lake's* pressure gauge at 125 feet showed fifty-two pounds to the square inch. It was 7 minutes and 15 seconds before her coning tower disappeared under the water, and from the time the *Lake* was thirty feet beneath the surface till she reached the bottom it was only 3 minutes and 45 seconds. In all it required the boat 19 minutes and 18 seconds from the time she was sealed to sink to the bottom, and altogether the boat was sealed about 20 minutes and 48 seconds. After she came to the surface a hurried examination showed that no water had leaked into the *Lake,* which was verified in a more careful examination that was made later. The boat's

machinery had also escaped damage, for under power of her gas engines she ran a mile into her slip at the torpedo station. A model of the Burger subsurface boat was also given a trial by the Board. She made three runs over the measured-mile course in the bay, the first at a rate of 8.93 knots, the second at a rate of 8.59 knots and the third at a rate of 9 knots, the mean being 8.78 knots. The boat is only 35 feet on the water line and is driven by a 28-horsepower gasoline engine.—"Army and Navy Journal."

ENGLAND.

Afridi.—The torpedo-boat destroyer *Afridi,* together with her sister ship *Ghurka*, will constitute the fastest class of vessel in the British Navy. The *Afridi* is an ocean-going torpedo-boat destroyer, was launched May 8th, from the Elswick shipyard of Sir W. G. Armstrong, Whitworth & Co., Limited. Her length between perpendiculars is 250 feet, and her breadth, molded, 25 feet. Her depth, molded, is 15 feet 6 inches, and her mean draught 7 feet 1 inch. The ship will carry three 12-pounder quick-firing guns, two of which will be mounted on the forecastle deck, and one on the upper deck aft. She will also be fitted with two 18-inch torpedo tubes, which will be mounted on the upper deck.

The machinery will be supplied by the Parsons Marine Steam Turbine Company, Limited, of Wallsend-on-Tyne. It will consist of a set of compound turbines, which will drive three propeller shafts, each fitted with one propeller. The boilers will be of the Yarrow type, and will develop approximately 14,500 horsepower. It is interesting to note that the Admiralty have decided to use oil instead of coal in these ships. Both the *Afridi* and the *Ghurka* have been designed for a speed of 33 knots.

Ariel.—During the maneuvers at Malta, April 19, the British destroyer *Ariel* struck the breakwater at the entrance of Grand Harbor, broke in two and sank. One man was drowned.

The Cossack, an ocean-going torpedo-boat destroyer, built by Cammell, Laird & Co., Birkenhead, was launched February 16th. Dimensions: Length, 270 feet; beam, 26 feet. She

has Parsons turbines, with triple screws. The designed speed is 33 knots.

Launch of a New Cruiser.—The first-class armored cruiser *Defence* was successfully launched at Pembroke Dockyard on April 27th. The new cruiser is a sister ship to the *Minotaur,* which was launched from Devonport Dockyard in June last, and her distinctive feature is her armament, which will be mounted entirely on the upper deck or above it. She will have four 9.2 inch breech-loading guns mounted in pairs in barbettes situated one on the forecastle deck and the other on the upper deck aft, and both at the middle line, and ten 7.5 inch breech-loading guns mounted five on each side in barbettes at regular distances apart over the portion of the ship occupied by the boiler and engine rooms. The mounting of so many guns on the upper deck of the *Defence* necessitated the reduction of weight in the upper structure of the ship, and, accordingly, instead of the vertical side armor being carried upwards from the lower to the upper deck in the central part of the vessel, it is stopped at the main deck. The side armor forms a complete belt round the ship, and is six inches thick outside the engine and boiler rooms, tapering to four inches towards the bow, and to three inches towards the stern. The barbettes are formed of armor seven inches thick, and in addition to the guns referred to, the *Defence* will carry sixteen smaller quick-firing weapons. She has also been fitted with five submerged torpedo tubes, and her engines are designed to give a speed of twenty-three knots per hour. The cost of the *Defence,* according to the latest revised estimate, will be £1,362,970.

Sale of Obsolete War Vessels.—Several obsolete war vessels have been disposed of at the following prices: Battleship *Sans Pariel,* T. W. Ward, Sheffield, £26,600; battleship *Conqueror,* Castles' Shipbreaking Company, Ltd., London, £16,800; armored cruiser *Undaunted,* Harris Bros., Bristol and Falmouth, £14,400; torpedo gunboat *Alarm*, the Shipbreaking Company, Ltd., London, £3,650; steam yacht *Warc*. W. Thomas and Son, shipbreakers, Anglesea, £925; torpedo-boat destroyer *Skatė,* Cox & Co., Falmouth, £305.

Warships at present under Construction in the United Kingdom.

Nationality.	Description.	At Royal Dockyards. Yard.	No.	Displ'm't.	At Private Yards. Yard.	No.	Displ'm't.	Total. No.	Displ'm't.
				Tons.			Tons.		Tons.
British	First class battleships	Portsmouth	1	18,600	Elswick	1	18,600	5	88,800
		Devonport	1	18,600	Dalmuir	1	16,500		
					Jarrow	1	16,500		
	First class armored cruisers	Chatham	1	14,600	Clydebank	1	17,250	7	109,100
		Devonport	1	14,600	Elswick	2	30,800		
		Pembroke	1	14,600	Govan	1	17,250		
	Torpedo-boat destroyers	....	....	....	Birkenhead (2), Cowes (1), Elswick (1), Hebburn (1), Woolston (2)	7	6,585	7	6,585
	First class torpedo boats	....	....	....	Chiswick (2), Cowes (6), Hebburn (2), Poplar (3), Woolston (2)	15	3,725	15	3,725
	Submarine boats	Chatham	2	630	Barrow	12	3,720	14	4,350
	Total	....	7	81,630	....	41	130,930	48	212,560
Foreign or not stated	Battleships	....	....	....	Elswick	1	19,000	1	19,000
	Cruisers	....	....	....	Barrow	1	15,200	1	15,200
	Torpedo-boat destroyers	....	....	....	Jarrow	2	740	5	1,760
					Poplar	3	1,020		
	Total	....	....	....	....	7	35,960	7	35,960
Total		....	7	81,630	....	48	166,890	55	248,520

—"Lloyd's Quarterly Shipbuilding Return."

Launch of H. M. S. "Invincible."— The new cruiser *Invincible* was launched from the Elswick yard of Sir W. G. Armstrong, Whitworth & Co., Limited, April 13th. Complete success attended the event, which is as one would expect, inasmuch as there have already been floated from the same establishment, since it was first inaugurated 22½ years ago, twenty-four fighting ships for the British Navy, apart altogether from the work done for foreign Powers.

The *Invincible* excels, alike in length, speed, and gun power, any of her predecessors. She has a length of 530 feet, a beam of 78 feet 6 inches, and when displacing 17,250 tons draws 26 feet of water. At this draught she has a capacity for carrying 1,000 tons of coal, and a large quantity of oil fuel. While protected as effectively as the majority of battleships, excepting perhaps only the *King Edward VII* and *Dreadnought* classes, she is excelled in her gun power only by the *Dreadnought.* The main armament includes four pairs of 12-inch guns, mounted in four barbettes—one forward, one aft, and two amidships, the latter *en échelon* between the second and third funnels, which are, therefore, at a greater distance apart than the two forward funnels. The magazines for these guns are located between the second and third boiler-rooms. This arrangement of guns will enable the amidship guns to fire on either broadside, although their arc of training forward and abaft of the beam on the off broadside may be more limited than in the case of the *Dreadnought.* In the broadside action, however, the ship will have the same fire as the *Dreadnought,* while forward as well as aft she will use six 12-inch guns. The secondary armament for repelling torpedo attack consists of a large number of 4-inch quick-firing guns. The turbine machinery, which is being constructed by Messrs. Humphrys, Tennant & Co., of London, is to develop 41,000 indicated horsepower, and the legend speed is 25 knots. There are four screws and two rudders. All of the screws have four blades, and the outer propellers are about 30 feet ahead of the inner propellers, the latter being close up against

the rudders, which are entirely suspended from the counter of the ship within which the steering gear is fitted.

The vessel is in an advanced state, the launching weight being 8,400 tons. The drags consisted of nine piles of chains, resting on the ground on each side of the ship, the total weight being 450 tons. The cradle, which in the case of fine-ended ships is always a subject of interest, was made up somewhat differently from the practice in most naval construction works. The rows of poppets at the forward end were not continuous. There were three spaces left. The tops of the poppets were housed in the steel structure, from which there were plates extending under the ship, from the port to the starboard poppets; but instead of this metal structure, with its lashings, being secured to the hull, as in some instances, it was independent, and the ship was snugly cradled inside the plating with timber wedges. In this way some play was possible within the cradle, a matter of some importance when the stern of the ship is floated in the process of launching. It has been found in recent launching practice that, in cases where the framing has been bossed out for twin-screw propeller shafts, the stern becomes waterborne at an earlier period than was formerly the case, and, as a result, the consequent thrust on the forward cradle is earlier and the tendency to movement of the ship of greater duration. There is, however, the further fact that the speed down the ways does not attain the same high velocity. In the case of the *Invincible* the time occupied from the first perceptible movement until the vessel was completely afloat was just under 50 seconds. It follows that the distance which the vessel traveled from the end of the ways is shorter.

The release of a vessel of such weight is also of interest, and the method adopted at the Elswick Works was notably successful. The last dagger dropped was hydraulically operated. This dagger was of steel, pivoted to the bottom of the standing-way, and projecting through it into a bearing constructed of forged steel in the bottom of the sliding-way. To the upper part there was secured a counterbalance weight,

while the bottom abutted against a ram working in a cylinder in which the hydraulic pressure was 100 pounds. Lady Allendale, who released and named the ship, working a small capstan, opened the valve, and the receding of the arm within the cylinder caused the dagger to incline to the horizontal position, and thus gradually released the sliding ways and the ship. The ship was held rigidly until the moment of release, and at the same time there was certainty of release. Indeed, the whole performance was characterized by precision; and at the luncheon there was general congratulation upon the success of the day's proceedings, and upon the prosperity of the Elswick Works under Sir Andrew Noble, who presided.—"Engineering."

Launch of H. M. S. "Indomitable."—On Saturday, March 16th, his Majesty's ship *Indomitable* was successfully launched into the Clyde by the Fairfield Shipbuilding and Engineering Company, Limited, Govan. The launch was to have taken place at a quarter past two o'clock, but the tide proving especially good, it was found expedient and advisable to float the vessel a little before the pre-arranged time, and in this way a goodly proportion of the intending spectators were disappointed in not witnessing the actual send-off. The religious service usual at the launch of vessels for the British Navy was conducted on the special platform at the bow of the ship by the Rev. Rodger S. Kirkpatrick, minister of Govan Parish, and at four minutes past two o'clock the Marchioness of Breadaldane, who had been requested by the Admiralty to perform the ceremony of naming and releasing the vessel, cut, with an ornamental hatchet, the cord which controlled the arrangements of release. The customary bottle of wine was smashed against the ram—a feature which is much less pronounced than in older types of warships—and a moment later the vessel began to move down the ways. From the first movement until the hull became completely waterborne the time occupied was 1 minute 20 seconds. The tugs in attendance smartly took the *Indomitable* in charge, and she was soon berthed in the company's fitting-out basin.

On account of the Admiralty having made strict stipulations to that effect, no official description of this latest and mightiest of British cruisers has been furnished by the Fairfield Company. So faithfully has this desire for secrecy been regarded that amongst the workmen, and, indeed, all interested in naval matters on the Clyde as well as over a wider area, the *Indomitable* has come to be spoken of as "the mystery ship." The class of which she is a unit has excited world-wide interest, and although the details of design have not been officially disclosed, the principal particulars are not unknown to those well informed on naval subjects. With due reservation, what follows regarding the Fairfield cruiser and her two sister ships—the *Inflexible*, on the stock at Clydebank, to be launched on March 30th, and the *Invincible*, being built at Elswick, to be launched on April 13th, may be taken as in the main correct.

The dimensions, power and speed, etc., of the three ships, which are identical in design, are: Length, 530 feet; breadth, 78 feet 6 inches; draught, 26 feet; weight of hull, 9,660 tons; displacement, 17,250 tons; indicated horsepower, 41,000; speed, 25 knots; coal capacity at load draught, 1,000 tons. The vessels have a complete broadside armor from bow to stern, the maximum thickness being 7 inches, tapering to 4 inches at each end. The armor extends from under the water line to near the upper deck. In respect of protection, therefore, the *Indomitable* and her consorts are, perhaps, superior to previous vessels, just as they are certainly superior to their predecessors in gun power. Indeed, this new class of cruiser might, almost under any condition, tackle heavy battleships. Their length is 30 feet greater than that of any preceding British cruiser, and the displacement tonnage is 2.650 tons more than that of any hitherto built. The Fairfield-built cruiser *Cochrane*, for instance, which has recently been put through all her trials with conspicuous success, is 480 feet long as against the *Indomitable's* 530 feet, has 13,550 tons displacement as compared with the 17,250 tons, and 23,500 indicated horsepower as compared with 41,000 indicated horse-

power. The designed speed of the *Cochrane* was 23 knots as compared with the 25 knots of the *Indomitable*.

The machinery of the three vessels is of the Parsons steam-turbine type, steam being supplied from water-tube boilers, some of the vessels having the Yarrow boiler, and others the Babcock & Wilcox. As in the case of the battleship *Dreadnought,* the new cruisers have four lines of shafting, each carrying a three-bladed propeller of bronze. In the disposition of propellers, and in the design of the after run of the vessel, it was seen, from distant observation of the vessel before her send-off, that considerable departure from conventional practice has been made. The general impression conveyed was of effort and ingenuity having been exercised in diminishing needless excrescence and surface in the form of "deadwood," and leaving everything conducive to solid and undisturbed water in the region of the propellers; the rudder apparently being hung or supported on a very light structural arrangement. On each of the two inner shafts there will be a cruising turbine, a low-pressure ahead and a low-pressure astern turbine, while on each of the outer shafts there will be a high-pressure astern turbine, as well, of course, as the forward turbine. For low powers the steam thus passes through three turbines before entering the condenser, and the range of expansion is suggested by the fact that the blades of the turbines vary in length from something less than 1 inch up to 16 inches. There is a longitudinal bulkhead dividing the port and starboard engine rooms, in each of which there are thus five turbines and a condenser.

Even more than speed, the dominant feature of the *Indomitable* and her consorts is armament. The advance in this respect over previous vessels of the kind is enormous. Compared, for example, with the first-class cruiser *Cochrane*, recently handed over by Fairfield, the *Indomitable* has over double the hitting energy. The *Cochrane* has six 9.2-inch and four 7.5-inch guns, equal to a force of 161,470 foot-tons for one full broadside round. In the *Indomitable* there are eight 12-inch guns, equal to the enormous destructive power

of 381,576 foot-tons for one broadside round. This forms a striking example of the advance made within a period of two years in the aggressive or defensive powers of a type of vessel in which at first high speed was the distinctive quality aimed at. In the new vessels there is a corresponding gain in bow and stern fire, as they can operate six 12-inch guns ahead and six astern.

The progress made with these ships has been very satisfactory. The keel of the Fairfield vessel was laid on March 1st, 1906, and by the end of this March she will have had spent upon her the sum of £1,470,375. The keel of the Barrow ship, the *Inflexible,* was laid on February 5, 1906, and has had voted for progress with her construction up till the end of March the sum of £1,421,479. The keel of the *Invincible* at Elswick was laid on April 2d, and on her there will have been spent up to the end of March, £1,417,732. There is every prospect of all the vessels being completed for service fourteen or fifteen months hence, well within the contract time of two years and three months.—"The Engineer."

New Destroyers.—There was successfully launched from the works of Messrs. John Samuel White & Co., Ltd., East Cowes, Isle of Wight, on March 15, H. M. S. *Mohawk,* one of the new ocean-going destroyers building under the naval program of 1905–6. Her length is 270 feet, with a displacement of 800 tons. She will have an armament consisting of three 12-pounder quick-firing guns, and three 18-inch revolving torpedo tubes. The speed, to be maintained on a six-hours' full-power trial, is 33 knots, the radius of action at economical speed being 1,500 nautical miles. The vessel will be propelled by turbine machinery comprising five turbines (three ahead and two astern), driving three shafts and propellers, built by Messrs. J. S. White & Co., Ltd., under license from The Parsons Marine Steam Turbine Company, Ltd. The power of the machinery aggregates about 14,500 I.H.P. Steam will be supplied by six boilers, each of about 2,400 H.P., of the "White-Forster" type, made by the same firm, these boilers being fired by liquid fuel on a system which has been

experimented with successfully by the British Admiralty for some two or three years. No coal stowage is provided in the vessel, and she will rely entirely on the liquid-fuel installation. The air pumps, forced-lubrication pumps, boiler-feed pumps, oil-fuel pumps, fire and bilge pumps, and distilling plant are being supplied by Messrs. G. &. J. Weir, Ltd., Glasgow, the fans and circulating engines by Messrs. W. H. Allen, Son & Co., Ltd., of Bedford, and the air compressing machinery and electric light plant by Mr. Peter Brotherhood, Westminster Bridge, London, S. E.

The Torpedo-boat "Grasshopper."—The launch of Torpedoboat No. 9 (lately known as H. M. S. *Grasshopper*) at the Chiswick Yard of Messrs. John I. Thornycroft and Co.. Ltd., on March 18, is an event of considerable interest from the fact that it is the fifth of the new class of British turbine-driven torpedo boats constructed at this yard, and in addition marks the end of the long and notable list of torpedo craft constructed by this firm at their Chiswick works, commencing with the *Lightning,* known in the Service as Torpedo Boat No. 1, which was delivered in 1877.

The *Lightning* was a boat of 85 feet in length, and capable of a speed of 18 knots, which was then considered most remarkable. Torpedo boat No. 9 is 168 feet long with a beam of 17 feet 6 inches and a draught, under fully loaded conditions, of 5 feet 11 inches. She is fitted with Parsons turbines and Thornycroft boilers using oil fuel, both the turbine machinery and boilers being constructed by Messrs. Thornycroft.

The contract speed is 26 knots, but in the case of the four earlier boats which have already undergone trials, this speed has been exceeded by considerably over a knot. The armament consists of two 12-pounder quick-firing guns and three torpedo tubes. Torpedo Boat No. 8, one of the sister vessels. having successfully carried out all her trials, was handed over to the Admiralty officials at Sheerness by Messrs. Thornycroft on Saturday last. Messrs. Thornycroft now propose to carry

out all work of this nature at their Woolston Works, Southampton, where they are much more favorably situated than at Chiswick.—"Page's Weekly," March 22, 1907.

FRANCE.

The French Naval Disaster.—While the French battleship *Jena* was lying in drydock at Toulon March 12th her powder magazines exploded, completely wrecking the vessel, killing 118 members of her crew, including the commanding officer, Captain Adigar, and Captain Vertier, Chief of Staff of the Mediterranean Squadron, and wounding thirty-eight others, including Admiral Macerou. A child was killed some distance away by an exploding shell from the ship, and many other civilians were injured.

There were three explosions in the interior of the ship, the first occurring at 1.30 P. M., and the others a few minutes later. As yet it has not been learned how the explosion was caused. One statement is that the torpedo men were manipulating a torpedo, which, under some shock, exploded. The violence was so great that two shells in the explosives magazine of 305 millimeters collided and exploded, starting the fire. The gases so produced being compressed in the magazine caused the other explosions. The explosives were hurled immense distances over the town and suburbs. The basin in which the *Jena* was lying to be careened was dry and when the explosion occurred it became imperative to flood the dock to quench the fire aboard and prevent further explosions. The gate was blown in by a shell from one of the warships present.

The *Jena* had just undergone a final inspection of her hull and machinery, the latter having been completely overhauled, preparatory to joining the squadron on March 13th. The crew was in its full strength, being composed of the rear admiral, twenty-four other officers, and 630 men. The magazines had been replenished recently, and contained many tons of both smokeless and black powder as well as a number of charges for torpedoes. The crew of the *Jena* had finished

their mid-day meal only a short time before the explosion, and had dispersed to various parts of the vessel. Most of the men were engaged in work connected with the approaching departure of the ship, but quite a large party had been detailed to attend a lecture that was being given forward.

The French Minister of Marine visited the wreck on March 13th, after the drydock had been emptied of water and the vessel was resting on timber supports. She is not damaged forward, but aft she is terribly injured. There is one especially great rent in the hull. This rent proved a safeguard, as through it the gases caused by the explosion escaped, and thus the vessel was saved from utter destruction. The propellers and helm are undamaged. On board, on the various decks, everything was destroyed. Piles of earthenware mark the dining places, and the débris of woodwork indicates the sites of the cabins. The metal planks resisted the fire, although they are warped and displaced.

Admiral Macerou owes his escape from the fate of other officers, who were burned in their cabins through the blocking of their doors by débris, to the fact that the port holes of his cabin were open when the explosion came.

Various theories as to the cause of the disaster have been advanced. One is that the first explosion was caused by an electrical short circuit, but this is regarded as untenable, and so also is the theory that it was due to the ignition of a shell loaded with black powder. It is thought by many that the accident was due to the spontaneous explosion of smokeless powder. This powder is supposed to be tested every six months, but inquiries among surviving officers fail to establish whether this was done or not on the *Jena*.

Lieutenant Tiercelin, an officer of the *Jena* who was seriously wounded, says in a statement to the Minister of Marine: "I do not believe that the careless handling of anything produced the explosion, for all the men at work were old quartermasters. They were testing the ammunition elevators in the magazine of the 100-millimeter guns. The magazine was under the electric-lighting apparatus, which was below the

officers' quarters, and this fact explains the great number of officers killed. I do not believe that a short circuit was the cause, but solely the age of the powder and its fermentation."

The Toulon correspondent of the "London Daily Mail" quotes a French navy officer as saying that had the accident occurred at sea the vessel must have foundered instantly. Naval experts are astonished by the fierceness with which the flames burst out and raged furiously in the interior immediately after the explosion. That a battleship of modern construction should blaze like a grease box from an explosion arouses the gravest question in reference to naval construction. The correspondent describes the vessel as being as battered and torn as though she had fought a fierce battle. From the forward turret to the stern gallery of the ship is one blistered, riven mass of tortured steel and miscellaneous ruin. Twenty-four hours after the disaster, despite the fact that oceans of water had been pumped between decks, the fire had burned right through, leaving only one-tenth of the entire length of the vessel unharmed. She is almost completely burned inside and out to the very summit of her fore signal-mast. The fire consumed all the boats, ate away the deck planking, destroyed every particle of furniture in the officers' and men's quarters, ruined more than ten guns and seriously injured even the heavier ones. Only the bow guns are undamaged.

Inspector-General Admiral Blenaimé has expressed the opinion that the explosion was caused by the spontaneous combustion of powder, which disintegrates after a certain time and which requires constant inspection for safety. Spontaneous combustion was the cause of the terrible explosion of the *Toulon* powder magazine in 1899.

The *Jena* is a battleship of 12,052 tons displacement, 400 feet in length, sixty-eight foot beams and maximum draught of twenty-seven feet six inches. She was laid down at Brest, January 15, 1898, has three screws, two turrets, two funnels, and has three sets of engines capable of developing 15,500

horsepower and a speed rate of eighteen knots. Her armament consists in part of four 12-inch guns and four torpedo tubes, and her complement number 630.

The destruction of the *Jena* will command the serious attention of naval scientists throughout the world. It is another reminder that modern warships embody elements of construction and the use of materials which are not yet freely understood and which require the most careful study in the interest of safety in peace and of effectiveness in war. The high explosives employed in these great vessels seem capable of undergoing certain chemical changes which make them extremely dangerous even with the most careful handling. The mysterious sinking of Togo's battleship *Mikasa* at Sasebo has been officially attributed to an explosion caused by such changes in her ammunition, and there appears good reason to believe that the destruction of the *Jena* was due to similar causes. Besides these there have been in recent years many other but less serious accidents on large warships in various navies, the causes of which have never been clearly explained—all going to show that the latent properties of the high explosives now in use are not thoroughly known.

Increase in French Submarines.—There are two classes of sub-surface boats in the French Navy, the lines between being well defined; the submarine class is for strictly harbor defense purposes, while the submergible class is rated as seagoing, thus protecting within a greater radius. The two schools have had separate exponents in the French marine, and, except in the cases of *X* and *Z*, there has been no modification from one to the other; these units were submergibles, designed by engineers who had theretofore created submarines, and are regarded as experiments not promising well. There is no example in the opposite direction.

According to the latest information, a total of eighty-nine units have been authorized to be built by the French, of which thirty-one were submarines and the remainder, fifty-eight, submergibles.

The submarines were first in the field with one unit in

1888 and again one in 1893. In 1899 the first "type" appeared, the *Morse,* of which in 1901 two units were finished. In 1901-1902 the ill-fated *Farfadet* type came out with four units. Of the former type one recently sunk, unmanned, and of the latter two units were lost with all on board.

In 1903 the *Perle* type began to appear, and the proposed series of twenty units is not yet complete, it being apparently in question as to completion; this type was based on the *Gustave Zédé* of 1893. Finally there are two units of the small *Guêpe* type building—small reproductions of the *Farfadet* type.

The pioneer designer of sub-surface boats in the French Navy was Gustave Zédé, who designed the submarine *Gymnote;* he was followed by E. Romazotti with a submarine named for his predecessor, the *Gustave Zédé;* later came his *Morse.* Engineer Maugas designed the *Farfadet* type, and lastly M. Petithomme the *Guêpe* type. As the pioneer boat is out of service, and the little *Guêpes* are not yet finished, it may be said that the entire submarine class in the French Navy today is built on the designs of Romazotti and Maugas. It may be noted that each of these designers produced one submergible experiment, neither of which has been duplicated up to the present time, they being the *X* and *Z* mentioned earlier in this summary. Though, and perhaps because of, coming later, the submergible class is much more numerous in the French Navy than the submarine class. In 1899 Engineer Laubeuf produced the *Narval,* which has been the prototype of the entire class of French submergibles; in 1901 four more units of this type were launched.

In 1904 the *Aigrette* type appeared—two units—and there are now building in the various French naval stations twenty-one of this design, with three more series, aggregating twenty units, projected and included in the totals cited above. In 1905 Bertin designed an experimental submergible along Laubeuf lines, and also the *Omega,* which is styled an "official development" of the Laubeuf type; on this *Omega* type there are now being constructed six units, known as the *Emeraude*

type, making a total of Laubeuf submergibles of fifty-six out of a total in the class of fifty-eight, the other two being the Romazotti and Maugas experiments.

It may be justly assumed that the submergible class is preferred in France from the fact that of eighty-nine existing and projected sub-surface units fifty-eight are submergibles and thirty-one submarines—an even greater disparity promises, from recent developments, to obtain in the future; so far no disaster has ever occurred in one of the submergible class. The development of sub-surface boats in France has outstripped numerically that of any other country many times over; in the United States, for instance, we have in all twelve units, including those at present under order—less than one-seventh of the number of the French, with many times the number of harbors to defend. The curve of displacement, chronologically considered, has been extremely irregular in the submarine class, while in the submergible class there has been a steady increase in tonnage, as will be seen from the appended condensed table:

Submarines.	Tons.	Submergibles.	Tons.
1888, *Gymnote*, . .	30	1899, *Narval*, . . .	200
1893, *Gustave Zédé*, .	270	1904, *Aigrette*, . .	250
1899, *Morse*, . . .	146	1905, *Omega*, . . .	375
1901, *Farfadet*, . .	185	Bldg., *Emeraude*, . .	430
1903, *Perle*, . . .	68	" *Q*," 53–60, .	500
Bldg. *Guêpe*, . . .	44	Projected, ...	800

Since the data was collected on which the above statement is based Chief Engineer Laubeuf has resigned from the French Navy, and the present Minister is now proceeding to build several units of largely increased tonnage in the submergible class,

ITALY.

The Italian Navy.—The Italian naval authorities have decided to sell or break up a number of battleships and other vessels during the five years 1907-12. The list comprises 21 ships of various classes, including the *Duillio* (launched in 1877) and the *Andrea Doria* (launched in 1891), as well as

51 torpedo boats. With the proceeds, which are estimated at £250,000, the Italian Government intends to make large purchases of coal for its navy.

Submersible Boat "Glauco."—The *Glauco* is the first of a series of five submersible torpedo boats which the Italian Navy will have completed in the course of a few months. Their displacement when fully emerged is 175 tons, and 220 tons when completely submerged. They are designed for a speed afloat of 15 knots, using the explosion motors only, and for a cruising speed of 10 knots. The fuel carried is sufficient for a run of 175 miles at the higher speed, and for 600 miles at the 10-knot speed. The time required for diving when fully emerged is five minutes. Each boat carries two torpedo tubes for 18-inch torpedoes. Their length and breadth at waterline afloat are, respectively, 42 meters and 4.3 meters (137 feet 9 inches and 14 feet 11 inches. The reserve of buoyancy when fully emerged is said to be 120 tons.

Engineer C. Laurenti is now Technical Director of the Fiat-Muggiano Company, Spezia, which, we are informed, has in hand at the present time several submarine boats embodying improvements on the *Glauco* type.

ROUMANIA.

Torpedo Vedette Boats.—Eight screw torpedo vedette boats have recently been built for the Royal Roumanian Government by the Thomas Iron Works, Shipbuilding and Engineering Company, London. The vessels are 100 feet long, 13 feet beam, with a draught of water of 2 feet 9½ inches, and a displacement of 51 tons.

The hulls are built throughout of steel, the deck and sides to just below the water line being bullet-proof. The vessels are each fitted with a 47-millimeter gun on top of the conning tower forward, and a small mitrailleuse gun aft. Gear for dropping torpedoes over the side is fitted on both sides amidships, and two spar torpedoes are fitted forward over the bows. The boats are also supplied with powerful search-

light projectors, and are lighted throughout by electricity. Accommodation is provided for four officers and twelve men.

The problem set the Thames Company was an extremely difficult one, the maximum length allowed being 100 feet; maximum draught, 2 feet 9½ inches, and the speed 18 knots on a trial of four hours duration, under penalties. Owing to the abundant supply of oil in Roumania it was also desired to use petrol residuum as fuel. It was found impossible to use internal-combustion engines on account of their excessive weight, so it became necessary to provide the 600 indicated horsepower on a weight of 16 tons by using steam propelling machinery.

There are two sets of compound engines of the inverted type, the twin screws running in tunnels formed in the stern of the boat. The cylinders are 8½ inches, and 17 inches in diameter respectively; with a stroke, 9 inches, and they are supported on turned-steel columns strongly braced together. The main bearing frames are separate castings of Stone's high-tension bronze of 30 tons per square inch tensile strength, bolted to long steel girders, which also carry the thrust block, feed pumps and air pumps. The bearings throughout are lined with white metal, and centrifugal lubricators are fitted to all the revolving parts. The crank shafts are solid, and balance weights are fitted to the cranks.

There are two feed pumps fitted to each main engine, these being worked from a countershaft driven by worm and worm wheel from the crank shaft at a reduced speed, the pumps making one revolution to four of the engines. Each pump has a separate connection to the boiler through two non-return valves, and hand-controlled valves are fitted on the suction side only. This method of working is very simple, and is found to render the feeding of the boiler very safe and easy. There is also no danger of burst feed pipes owing to feed valves being inadvertently closed. The air pumps are driven off the forward end of the crank shaft through a universal joint, and run at the same speed as the main engine. Each set of machinery is complete with its own condenser, and the centrifugal circu-

lating pump, steam, exhaust, feed and other pipes are all in duplicate, so that each main engine is entirely independent of the other.

The circulating pump has an impeller 12 inches in diameter, driven by a single-cylinder engine. The pumps are entirely of gun metal, and from each pump there is driven an auxiliary air pump, which is so fitted that it can be stopped whilst the main air pump is working. The circulating water passes once through the condensers, and the condensers and pipes are arranged so that, with the circulating pumps stopped, enough water will pass through the condensers to maintain a vacuum of 22 inches, or 23 inches. There is a hot-well tank at the forward end of the engine room, and a reserve feed tank at the after end, both being provided with filters. An auxiliary feed pump, of the direct-acting type, by Messrs. Caird & Rayner, is fitted in the engine room. It has a cylinder 7½ inches in diameter, and a pump 5½ inches in diameter, with a stroke of 8 inches. This is large enough to feed the boiler at full power, and will do duty for bilge or deck, and will pump through the filter on the reserve tank. The propellers are of bronze, 3 feet 3 inches in diameter, three-bladed, accurately pitched and balanced. Steam is supplied by a Thames Iron Works water-tube boiler. The steam pressure is 185 pounds per square inch, and the heating surface 1,290 square feet. The furnace is specially arranged for burning oil fuel, the ordinary fire bars being replaced by fire brick, with slots for the admission of air. The casing is carried down under the hearth, and the air is admitted directly at the front, and at the back through a casing which serves the double purpose of keeping the back cool and of heating up the air before entering the furnace.

Owing to the very limited weight available for the machinery, it was necessary to resort to steam spraying. Four special burners are fitted to each boiler, two at the top having an angle downwards, and two on the bottom, at each side of the fire door, directed towards the center.

Oilbunkers are fitted on each side of the boiler, and a steam pump is fitted for pumping from them to the present-use tank.

A steam coil is fitted in each tank for heating the oil, the steam running first through the present-use tank, and then through the coils in the bunkers. The water from these coils is run into the bilge to obviate any danger of getting oil into the feed water. Both bunkers and the present-use tank act as settling tanks, and a valve is fitted at the bottom of each to drain off water as it collects. There is not enough height in the boat to run the oil through the burners by gravity; the present-use tank is therefore worked under an air pressure of from 1 pound to 10 pounds, the air being supplied by a small pump driven off the fan engine, and a reducing valve regulates the pressure.

The forced-draught fan is 4 feet in diameter, driven by a single steam engine running at 800 to 1,000 revolutions per minute, the air pressure at full power being from 4 inches to 5 inches of water. For the purpose of securing good ventilation in the engine room the air inlet of the fan is through the upper part of the engine-room bulkhead, and the mushroom-topped ventilator at the aft end of the engine room. The air for the boilers has thus to pass right through the engine room, serving admirably to ventilate that compartment. In the steering compartment is the generating set for lighting the ship and working the projector. The 8-kilowatt dynamo was made by the Thames Iron Works electrical department, and the enclosed tandem compound engine, made by Messrs. Peter Brotherhood, Belvedere road, Lambeth, S. E., has forced lubrication.

Four bilge ejectors, each of 8 tons per hour capacity, are fitted, one in each compartment. Chadburn's reply telegraphs are fitted from conning tower to both engines.

The boats were designed by the late Mr. G. C. Mackrow, and the official trial of the first boat took place on the Thames, September 25, 1906. The Roumanian Government were represented by Major Demetriade, Director of Marine; Captain-Commander Balesco, President of the Commission; Captain Negru, Captain Mihail and Captain Stefanesso.

The trial was of four hours' duration, and the following results were obtained:

Mean speed, knots,	**18.0365**
indicated horsepower	**622.7**
revolutions per minute	**554.8**
Boiler steam pressure, pounds per square inch	**182.3**
Air pressure in stokehold, inches water	**4½**
Pressure of steam in oil-fuel burners, pounds per square inch	**181**
oil for oil-fuel burners, pounds per square inch	**7**
Temperature of oil in warming tank, degrees Fahrenheit	**175**
Oil consumption, pounds per I.H.P. per hour	**1.73**

This result was obtained with the two top burners only in operation. The power was maintained steadily, and generally there was absolutely no smoke or visible gases from the funnels.

The machinery, which was designed by Mr. R. Warriner, ran smoothly and without vibration; no sign of heating was observed in any of the bearings, although only the ordinary oil service was used. On the stopping and starting trial the engines were easily and rapidly manipulated by one man. Four of these boats left the Thames Iron Works last October, taking altogether six weeks on their journey from London to Galatz through the waterways of Central Europe. They were not allowed to proceed under their own steam, but had to be towed along with the barges, etc., in the usual way. The last four boats left the works in March, and are now on their way through the center of Europe.

PERU.

New Peruvian Warships.—The *Coronel Bolognesi,* with her sister ship *Almirante Grau,* are the pioneers of the modern Navy which the Peruvians desire, and the latter ship had just returned from her trials when the *Coronel Bolognesi* was launched. These two cruisers are somewhat similar in design to the British scout class, although their armament will not enable them to stand up against heavily armored vessels. In combination with high speed, it is to enable them to harass heavy fighting ships, and at the same time to meet attack by torpedo boats.

The trials of the *Almirante Grau,* which have been con-

cluded, were arranged to insure that the vessel should be able to steam at sea under all weather conditions at twenty-two knots, to make when necessity arises twenty-four, and to be able to cruise at a low speed with a light consumption of fuel. On the trials on the Clyde the vessel averaged, on full power, over twenty-four knots with the boilers working under normal conditions; the coal consumption was lower than the average of the eight new scouts for the British Navy; and no difficulty was found in realizing twenty-two knots with only eight of the ten boilers.

The trials occupied a week on the Clyde. The first test was a series of progressive speed trials over the measured mile at Skelmorlie, and the results were up to expectations. On the measured mile she steamed 24.64 knots, and considerably exceeded twenty-four knots throughout the full-power trial, the mean power being 14,144, while the revolutions were 216 per minute. The Admiralty "constant" was 229. The continuous sea-speed trial was equally satisfactory. The Peruvian Government desired that this trial of twenty-four hours' duration should be divided into four successive periods of six hours at progressive powers, viz: 900 horsepower for the first six hours, 3,600 for the second six hours, 6,000 for the third six hours, and 8,500 for the fourth. The coal consumption at low powers is, of course, higher, because the proportion of steam to be generated for the auxiliary machinery is greater than on a high-power trial. At low power more than one-third is needed for auxiliaries, whereas on a full-power run the proportion may be less than one-tenth. The coal consumption ranged between 2.27 pounds per horsepower for the low power to 1.635 pounds for the high power—all within the guaranteed conditions. This coal-consumption trial was completed at Barrow.

The *Coronel Bolognesi*, on trial, made 24.726 knots, with the engines indicating 14,384 horsepower, at 218.7 revolutions per minute. The Admiralty "constant" was 228.

Each ship has a bow and stern-chasing gun of 6-inch caliber, with an arc of train of 270 degrees, and in addition there are

eight 14-pounder quick-firing guns, eight 1¼-pounder guns, with two submerged tubes for firing 18-inch torpedoes. She has for protection a deck of 1½-inch thickness extending the full length, and the conning tower is of 3-inch armor. The armor and armament have been made at the Vickers Works, at Sheffield.

The engines are of the twin-screw four-cylinder triple-expansion type, balanced on the Yarrow-Schlick-Tweedy system, and designed to develop 14,000 horsepower with steam at a working pressure of 250 pounds at the engines and 280 pounds at the boilers. There are ten water-tube boilers of the small-tube type, arranged in three separate watertight compartments, to work under forced draft on the closed-stokehold system.

Special care has been devoted to the natural and artificial ventilation of the ship, and to the heating, steam radiators being extensively employed. There is, of course, a complete installation of electric light. Although the ship is small, baths are provided in all the living quarters. Refrigerating plant and ice-making machinery, with cold-storage chambers adjoining, are notable features. There is also a mechanically-equipped bakery on board, so that the vessel excels many warship of much greater size in the elements conducing to comfort.

The *Almirante Grau* will be the flagship of the embryo Navy, until a more powerful ship is provided. Both cruisers are named after Peruvian heroes. The principal difference in the design of the two is that the *Almirante Grau* has a full poop to provide quarters for the admiral; in the *Coronel Bolognesi* there is only half a poop. Both ships have a length of 370 feet and a breadth of 40 feet 6 inches. The displacement of the *Coronel Bolognesi* is about 3,180 tons, with a draught of 14 feet 2 inches, and a block coefficient of 0.525; the *Grau* draws an inch more, and displaces 3,200 tons. The design called for 24 knots, with 14,000 horsepower.

MERCHANT SHIPS.

Japanese Cable-Steamer "Ogasawara Maru."—The cable-steamer *Ogasawara Maru,* which left Nagasaki on October 5th last for the cable-laying work between Tsushima and Kiushiu is the first ship of its class built in Japan. This steamer is owned by the Department of Communications of the Japanese Government, and was constructed at the Mitsubishi Dockyard and Engine Works, at Nagasaki. The design of the vessel was entrusted to Dr. C. Shiba and Mr. K. Suyehiro, professors of the Engineering College of the Imperial University of Tokio; the contract was placed June 1, 1905, and the work commenced in November of the same year; the vessel being launched on June 2, 1906, six months after the laying of the keel. She is a steel spar-decked twin-screw steamer of 1,455 tons gross, with engines developing 1,850 indicated horsepower at full speed. The hull and machinery were constructed under the Japanese Government's Shipbuilding Encouragement Law. The principal dimensions of the vessel are:

Length between perpendiculars, feet and inches	240–0
Breadth, molded, feet and inches	34–0
Depth, feet and inches	22–3

The guaranteed speed was 12 knots, the ordinary sea speed expected being 11 knots. During the official speed trials the maximum speed obtained was 13,286 knots, and the average of six runs between the two measured-mile posts, three nautical miles apart, was 13.085 knots.

The hull and machinery of the vessel were all constructed by the Mitsubishi Company, except the cable gear, which was supplied by Messrs Johnson & Phillips, of Charlton, Kent. Three cable tanks are provided in the vessel: the fore tank, of 20 feet 6 inches diameter, intended for the storage of shore-

end cables; main tank, 27 feet in diameter; and after tank, of 23 feet diameter. The total capacity of the three tanks gives storage for 600 tons in all of deep-sea cables. Each tank is provided with a cone at the center for storing battens or fresh water. The hatchways of the tanks are provided with girders, carrying bell-mouth crinolines, etc.

The structural arrangements provide two complete decks running fore and aft, and an inner bottom carried nearly the whole length of the vessel, a feature of the structure necessary in this class of vessel. The hull is subdivided into five water-tight compartments by transverse bulkheads.

The foremost cable tank, provided for the shore-end cables, is carried up to the level of the main deck. The main tank is brought 5 feet above the main deck, leaving ample clearance between the top edge of the tank and lower side of the deck above. The bottoms of these tanks rest on the top of the inner bottom plating. The after tank is similarly constructed to the main tank, the bottom, however, resting on the top of the shaft tunnel.

The vessel is rigged as a two-masted schooner. Besides the necessary cargo appliances, the foremast is provided with yards to facilitate the lowering of buoys, etc., in cable-laying work.

In front of the foremost cable tank, and on the main deck, the cable gear is located, as shown on the plan; this part of the deck is most carefully constructed for minimizing the vibrations caused by the motion of the cable engines.

The double combined picking-up and paying-out machine, which is placed forward of the foremost tank, is very similar to that fitted in the cable steamer *Pacific,* and was supplied by Messrs. Johnson & Phillips. The machine is fixed on the main deck, with the winding drums appearing through an opening in the spar deck, all the handles for controlling the machine being brought to the spar deck. This double machine practically comprises two complete and independent gears, both of which may be used either for picking up or paying out. This double gear is provided with two sets of steam engines, placed fore and aft of the gear; either of

the engines can drive the gear for picking up and also for paying out when the pull on the cable is moderate. When a heavy load is on the cable both engines are put into work conjointly.

The drums can be driven at three different speeds by changing the gear wheels by means of clutches, which action can be performed from the starting platform on the spar deck. In the ordinary cable-laying or picking-up jobs, one of the engines may be kept as spare, since the power of one only is always quite sufficient for ordinary working. This reduces to a minimum the risk of total disablement of the gear. The cable-drums are 6 feet in diameter by 21 inches between the flanges, and they have steel gearing teeth and brake rings bolted to the rims of the drums for driving and holding, the drums being arranged to run loose on the shaft. The main brakes encircling the rings attached to the drum consist of steel bands, with elm-block liners. They are made to close by means of right and left-handed screws actuated through worm gearing from handle wheels mounted on the stands on the starting platform. This arrangement gives very fine adjustment for the lighter load on the cable, and at the same time provides enormous holding power when necessary. To the back of each brake strap is fitted a water-service pipe, so as to cool the rubbing surface of the elm-blocks and drum. This cooling water is supplied from an independent small steam pump, which is arranged to be started or stopped from the starting platform. Both brakes can be coupled to each drum when an excessive load is on, and give a very powerful holding-back action to the machine.

The hauling-off and holding-back gears stand on the spar deck, and are driven from the geared drum by means of endless chain through a friction coupling and free wheel. These gears are made to traverse across the face of the drum by means of screws, in order to lead the cable to any position on the drum. There are three sheaves in the bow gear. They have U section, run loose on a fixed shaft, cast-iron whiskers or guards being provided between the various sheaves

and at the outsides to prevent the cable from getting out of the grooves. The stern gear has only one cast-iron sheave mounted on a girder fixed on the spar deck, and is also provided with whiskers. For registering the pull on the cable three dynamometers are provided, these being fitted with Messrs. Johnson & Phillips' patent arrangement of internal spring, and their maximum readings are 25 tons for picking-up purposes, and 5 tons for paying out. The usual cable leads of **V**-shaped sheaves, mounted on **A** frames, are provided fore-and-aft to the dynamometers.

The cable engines were tried under steam outside Nagasaki Harbor. The machine wound up the load of 25 tons, consisting of bundles of anchor cables, from the bottom of water 17 fathoms deep, and at the speed of one turn per 22 seconds, which is equivalent to 1.02 knots lifting speed of cable. In this case of heavy pull the two sets of engines were used conjointly. For the purpose of turning over the cable from one tank to another, or taking in cable from outside, two sets of electric hauling gears, driven by a 6-brake-horsepower motor of closed-in type, were supplied by Messrs. Johnson & Phillips; they are driven off the electric main on the spar deck.

The propelling machinery of this vessel consists of two sets of triple-expansion engines, constructed under the Japanese Government's Shipbuilding Regulations, 14¾ inches, 24 inches, and 40 inches in diameter, with a 27-inch stroke. The propellers are 11 feet in diameter, and have each three blades of manganese-bronze.

The steam-generating plant of this vessel consists of two single-ended Scotch boilers and one donkey boiler of the Cochran type. The main boilers are 14 feet in diameter and 11 feet 6 inches in length, with three furnaces of 3 feet 6 inches internal diameter for each. The flues are of Deighton's patent corrugated pattern, united to the combustion chamber with detachable ends. The working pressure of the main boilers is 185 pounds, and that of the donkey boiler 100 pounds per square inch.

One evaporator of 10 tons capacity, supplied by Messrs.

Caird & Rayner, of London, is fitted, with its accessories; it is so arranged that the steam produced may be admitted to the main and auxiliary condensers, and also to the intermediate-pressure receiver of the main engines.

The general-service and ballast pumps are of the usual Worthington type, and all of them are so connected that each is at any time able to send water to any of the cable tanks.

The electric installation of this ship consists of two sets of steam dynamos, supplied by Messrs. Clarke, Chapman & Co., of Newcastle-on-Tyne, working at 110 volts. A searchlight projector of 16,000 candlepower is fitted on the flying bridge. A refrigerating machine of the carbonic-anhydride system. supplied by J. & E. Hall, of England, is fitted in the port wing of the engine room, on the same seat as the steam dynamos. This machine is for cold storage and for manufacturing ice. A Kelvin sounding machine is provided, a James submarine sentry, and also a steam-driven sounding machine for lifting samples from the ocean-bed, as usual in cable steamers.

In boat equipment the steamer is provided with two sampans, two 22-foot steel life boats, and one steam cutter, 30 feet long. The steam cutter is of the Navy type, fitted with a water-tube boiler and one set of compound engines, capable of developing sufficient power to propel the boat at 7 knots.

As will be seen from the above description, the cable steamer *Ogasawara Maru* is constructed and equipped in a very efficient manner; she was officially tried under steam on July 30th last, on which occasion she went through her various tests to the complete satisfaction of the owners' representatives.—"Engineering."

Wreck of American Steamship "Dakota."—The great 20,714-ton American steamship *Dakota,* sister ship to the *Minnesota,* the largest ever built in America and among the largest in the world, struck on the reefs off the coast of Japan, 40 miles south of Yokohama, on March 2, 1907, and now lies partially submerged, one mile from shore. The passengers and crew were landed in safety, but their belongings were lost. The Japanese steamer *Omimaru,* which went to the assistance

of the disabled liner, returned to Yokohama, and reported being unable to get alongside the wreck owing to the heavy seas which were running.

The *Dakota* and her sister ship, the *Minnesota,* belong to the Great Northern S. S. Co., and operate on the line between Seattle and the Orient, calling at Yokohama, Japan, and Hong Kong, China, and occasionally making trips to Manila.

They were constructed by the Eastern Shipbuilding Co., at Groton, Conn., during the years 1903-4-5, the yard being erected expressly to build them. When they went into commission they were the largest—although not the longest—vessels in the world, a distinction which has since passed to the newer of the British-built trans-Atlantic lines. The *Dakota* is 622 feet between perpendiculars, 650 feet over all, 73½ feet beam, and 19 feet depth of hold, with a gross tonnage of 20,714 and net tonnage of 13,305. She is driven by twin screws, carried a crew of 253 and had accommodations for several thousand passengers, and an immense quantity of freight.

The *Dakota* cost $3,000,000 and is insured for $2,500,000.

Wreck of the "Berlin."—Early on the morning of February 21st, the twin screw steamship *Berlin,* belonging to the Great Eastern Railway Company, was wrecked at the mouth of the waterway leading to the landing station at the Hook of Holland. The *Berlin* was on her way from Harwich, and her voyage was nearly over when the disaster occurred. A full gale was blowing at the time and the waves were mountainous. The entrance to the waterways is between two jetties running out into the sea. These jetties are nearly parallel, and their line is from a little north of west to a little south of east. The navigable channel between them is not quite central, and though the actual water surface is some 2,250 feet wide, the deep water channel in which there is an average of nearly 30 feet, is in places not more than 300 feet in width. Elsewhere the water shoals, and in places is not more than 8 feet or 9 feet deep. The deep channel is nearer the north jetty than the south, and its direction is indicated to the navi-

gator during the day by means of buoys, while at night it may be followed by keeping two lights in line. The direction of the wind was such that the seas would have been beating right in between the two dams and against that on the north side. From some cause which has not been at present discovered, the ill-fated vessel on making for the channel struck the end of the north breakwater or dam and then turned broadside to the sea. Very soon she broke in half and the fore portion foundered, carrying with it numbers of the crew and passengers. The stern portion remained on the sand, and was continuously washed by enormous waves. Heroic efforts were made by the Dutch lifeboatmen and others, the Prince Consort himself taking an active part in the relief operations, but in spite of everything that could be done, only fifteen were brought to shore alive, and these in a pitiful plight. It is feared that nearly 130 persons lost their lives in this terrible accident—one of the worst which has visited Europe for many years.

OBITUARY.

CHARLES HAYNES HASWELL.

Charles Haynes Haswell, the oldest civil engineer of note in the world and one of the most distinguished marine engineering experts, died at his home in New York City, May 12, in his ninety-eighth year, his death being due to a shock attending a dislocation of the shoulder, which he had sustained the day before in slipping and falling on the floor of his house in West Seventy-eighth street. Up to the time of his accident he was in good health, and, although so nearly one hundred years of age, he was regularly at his work in charge of important construction operations for New York City.

Mr. Haswell was born of English parents in New York, May 22, 1809. At the age of 19 he finished his schooling, which had been along classical lines, and, having inclinations toward a technical career, he secured employment in the engine works of the pioneer establishment of James P. Allaire. While with this firm he was selected by the United States Navy Commissioners to take charge of the building of a steam war vessel for the Government and in 1836 joined the United States Navy as chief engineer, receiving his commission as engineer-in-chief in 1845. Some years prior to this date he conceived the idea of building a steam launch and in 1837 constructed the *Sweetheart*, which was the first private steam yacht launched. He held the rank of engineer-in-chief of the Navy until 1851, and during that period designed the machinery for ten war vessels and introduced numerous mechanical improvements for increasing the early steam navy's efficiency.

In 1847-48 he designed the entire boiler and engine equipment of the steam frigate *Powhatan*. Owing to a lack of professional aid and the urgent requirements of the service and

the contractors, he was compelled to personally design every detail and made the working drawings himself in the intervals between attention to the necessary duties of his office as engineer-in-chief. The design of the engines was novel in several respects and wholly so in the fact that they were set in wrought-iron frames, the first construction of its kind. The *Powhatan* had a length of 250 feet and a displacement of 3,600 tons. The engines were inclined and the two cylinders were 70 inches in diameter, the stroke being 10 feet. The engines were designed to run at 13.5 revolutions per minute. The steam pressure was 10 pounds per square inch, the indicated horsepower 1,100. The paddle wheels of this man-of-war of 60 years ago were 31 feet in diameter and the speed of the ship about 10 knots.

After retiring from the Navy in 1851 Mr. Haswell built several merchant steamships, and then engaged in engineering practice in New York. For over 40 years he was surveyor of steamships for the marine underwriters of New York. He designed and located the buildings on Hoffman Island in the lower bay and built the crib bulkhead at Hart's Island. He constructed what at the time were the most extensive and difficult building foundations in the city. For several years past he had supervised extensive work of construction and improvement at Riker's Island. From 1855 to 1858 he was a member of the board of councilmen of New York City, and during the latter year was its president. Later he entered the employ of the city, and during his many years of service acted as chief engineer of the dock department, superintending engineer to the department of charities and correction and the board of health, and consulting engineer to the board of public improvement. He was appointed by the latter body in 1898 to design and supervise the extension of Riker's Island, and in 1902 was appointed consulting engineer to the board of estimate and apportionment, which position he held at the time of his death.

Mr. Haswell first brought out his "pocket-book," which is now in its seventy-second edition, in 1843, the volume at that

time containing 284 pages. He was also the author of "Reminiscences of an Octogenarian," a book of memoirs covering the old New York from 1816 to 1860. He was a member of the Engineers' Club of New York and Philadelphia, and belonged to many engineering and scientific organizations, among these being the American Society of Mechanical Engineers, the American Society of Civil Engineers, American Society of Naval Engineers, the Institution of Civil Engineers and the Institution of Naval Architects of Great Britain.

WILLIAM DURELL STIVERS.

William Durell Stivers was born in Jersey City, N. J., February 20, 1871, received his education in the public schools, graduated from the High School in 1887, and pursued some special studies in mechanical engineering at the Cooper Institute in New York. He entered the DeLamater Iron Works in 1887, and was assigned to special service in the superintendent's office where he had unusual opportunity for acquiring a special training in shop management, engineering and experimental work. His great love for the profession—probably imparted to him in part by his uncle, the illustrious Chief Engineer B. F. Isherwood, U. S. N., with whom he was a great favorite and who took deep interest in his development—and his indefatigable diligence and interest in all mechanical pursuits caused him to make rapid strides, and he soon became useful in many ways during the various interesting experiments that were at all times conducted at the DeLamater Iron Works, as, for instance, those of the noted Ericsson expansion engine, the hot-air engine, the Belleville boiler, refrigerating and compressed air machinery and other constructions. After the dissolution of the DeLamater Iron Works in 1889, William Stivers entered the Quintard Iron Works as draughtsman, eventually rising to the position of Acting Superintendent. He supervised the building and installation of the machinery of the U. S. S. *Maine*, which was afterwards destroyed in Havana Harbor. He was the Works' representa-

tive on the trial trips of the U. S. S. *Concord, Bennington, Detroit* and *Marblehead.* He also had a prominent part in the trial trip of the U. S. S. *Bancroft.* In 1902 he left the Quintard Iron Works to accept the position of Superintendent of the Yonkers Works of the Otis Elevator Company, which position he held until March, 1904, when he was engaged by the C. W. Hunt Co. as Executive Engineer, remaining active in that place until his death, December, 1906. He was married in 1898 and leaves a widow.

He joined the American Society of Naval Engineers in 1904.

A. H. R.

CAPTAIN GEORGE H. KEARNEY, U. S. N.

By Commander John R. Edwards, U. S. N.

The death of this sterling officer was a distinct loss to the Navy. From the time he entered the Service, he commanded the respect of his seniors, the friendship of his messmates, and the obedience and esteem of his subordinates. During his whole career in the Navy he was regarded as an exceptionally strong man, whether viewed from the standpoint of engineering ability, general knowledge, mental attainments or unimpeachable moral character.

The fact that he was invariably assigned to important duty best tells of his professional esteem and ability. His character was such that there was ever manifest in his performance of duty those attributes and qualities which made him a strong example of the best traditions and spirit of the Service. His high sense of honor and splendid manhood fittingly supplemented his capacity of leadership. His reserve, combined with unselfishness of purpose, likewise prompted him to avoid rather than to seek honors that were his due, but in all matters pertaining to naval engineering policy and progress his judgment and experience were sought and followed.

But few officers possessed more moral courage than he, although he never placed himself in an attitude where it was necessary to exhibit such attribute until after earnest and mature consideration had been given to a subject.

His professional life-work is best reflected in the efficient and splendid manner in which engineering duty, both afloat and ashore, is now being performed by officers who were cadets at the Navy Academy when Captain Kearny was head of the Department of Marine Engineering and Naval Architecture at that institution. His character, integrity and manhood have been stamped upon those whose privilege it was to serve under him. The full measure of his worth as an officer and a man was best known to those to whom he was not reserved and to whom he could freely speak of his life's work and purpose.

BOOK REVIEWS.

INTERNAL COMBUSTION ENGINES. An Elementary Treatise on Gas, Gasoline, and Oil Engines, for the Instruction of Midshipmen. By JOHN K. BARTON, Commander, U. S. Navy. Size, 9⅛x5¾ inches; pages, 135; illustrations, 52; price, $1.50. Annapolis, Md., 1907, THE UNITED STATES NAVAL INSTITUTE.

Commander Barton has made a very timely and appropriate addition to the series of modern American text-books for use in the engineering department of the United States Naval Academy.

This book, however, is not to have its usefulness confined to the class rooms and experimental laboratories at Annapolis. It appears at a time when the field of application of internal-combustion motors is being rapidly and satisfactorily extended to marine purposes, chiefly as a result of the employment of hydrocarbon fuels which are removing many of the limitations hitherto imposed afloat.

In our own and in foreign services, extensive experimental research is underway and higher powers together with simpler and more compact installations are positively assured. Already many torpedo boats, submarines and tenders are operating with these motors.

The work at hand reviews the principal types of gas and oil engines in construction and operation.

The chapter on Gaseous Fuels and Producers is especially important since the marine producer is attracting most attention in the application of the motor to naval uses. The chapter on Gas Engine Tests renders the work thoroughly useful in the laboratory and is comprehensively arranged. It comprises the preparation of the data for and the procedure in an engine test, and includes, in tabular form, the report of such test and

also the report of a heat-balance test. The arrangement of apparatus for the determination of the calorific value of the fuel by means of the Junker Calorimeter is included.

The final chapter deals with the care, management and practical operation of the motors, with a mention of the troubles to look out for and how to remedy or avoid them.

The illustrations are good and the text clear—a boon to students.—W. T. C.

DIE SCHIFFSSCHRAUBE (The Marine Screw Propeller). By ALBERT ACHENBACH. Part II. Kiel. ROBERT CORDES.

Part I of this work, which is to comprise three parts, was noticed on page 360 of Volume XVIII. Part II, covering, with the appendix, 152 pages, with 20 plates, is devoted almost entirely to the construction of screws for both naval and merchant vessels. There are chapters devoted to the drafting, to material, and to the work in the foundry and machine shop, with tables of the dimensions of screws actually built for different classes of vessels; a separate chapter is given up to the Niki propeller. Although there is little that is new in this Part, the material is well presented and is comprehensive.

The appendix, of 40 pages and 5 plates, covers the screw arrangement in motor boats, and includes descriptions of couplings, feathering screws and reversing gear.

One disadvantage of publishing this work in Parts is apparent in Part II, where a few references are made to plates embraced in Part I.—R. S. G.

THE STEAM ENGINE AND OTHER HEAT MOTORS. By W. H. P. CREIGHTON, U. S. Navy (Retired), Professor of Mechanical Engineering, Tulane University of Louisiana. JOHN WILEY & SONS, New York.

This excellent work on the Steam Engine and Other Heat Motors, containing 500 pages, written particularly as a textbook, will be found of very great value to the practical engineer. Mr. Creighton, from his long experience, both as an

Engineer Officer in the Navy and as an instructor in mechanical engineering, is especially equipped for producing such a work as we have before us. The book is divided into eighteen chapters, with an appendix, as follows:

Chapter I.—Review of Elementary Principles and General View of Steam-Engine Plant.
II.—Steam-Engine Indicator and its Calibration.
III.—Curves and the Work of Expansion.
IV.—Zeuner and Bilgram Valve-Diagrams and Design of Plain Slide Valves.
V.—Measuring the Effects of Heat.
VI.—Measuring the Effects of Water and Heat on Water and Steam.
VII.—Measurement of Heat Losses.
VIII.—Entropy.
IX.—Condensers and Air Pumps.
X.—Small Auxiliaries.
XI.—Multiple-Expansion Engines.
XII.—Revolution Control.
XIII.—Speed-Variation Control.
XIV.—Steam-Engine Tests.
XV.—Superheated Steam and Steam Turbines.
XVI.—Gas Engines and Gas Producers.
XVII.—Boiling in a Vacuum.
XVIII.—Refrigeration.

Appendix.

Table I.—Properties of Familiar Substance.
II.—Hyperbolic or Naperion Logarithms.
III.—Heating Values of Various Substances.
IV.—Oxygen and Air Required Theoretically for the Combustion of Various Substances.
V.—Relative Humidity. Per cent.
VI.—Weights of Air, Vapor of Water, and Saturated Mixtures of Air and Vapor at Various Tempertures and Constant Pressure.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XIX. AUGUST, 1907. No. 8.

THE EXPERIMENTS MADE BY MR. UTHEMANN TO DISCOVER A PROCESS FOR PREVENTING THE CORROSION OF COPPER AND BRASS BY SEA-WATER UNDER THE CONDITIONS FOUND IN THE SURFACE-CONDENSERS OF MARINE STEAM-ENGINES.

Translated from "Le Génie Civil" of the 23d of September, 1905,

BY CHIEF ENGINEER B. F. ISHERWOOD, U. S. NAVY, MEMBER.

Mr. Uthemann, a Member of the Engineer Council of the German Navy, has recently made a series of experiments to discover a process that would prevent the corrosion of copper and of its alloys by sea-water. Before giving the results of his experiments, some facts relative to this problem may be recalled as germane to it.

The employment of copper-sheathing for the immersed exterior part of the hulls of wooden sea-going vessels dates back to 1761, when the British Admiralty, for the first time, commenced its use for preventing the adherence of barnacles and similar accretions to their hulls.

After the introduction of steam machinery into sea-going vessels, copper and its alloys soon superseded cast iron for the enormous quantity of piping required and for other purposes, on account of its less weight and bulk, properties of great value in that case, and which became more and more important as the power to be developed in the same sized vessels became greater and greater.

In proportion to the greatness of this development, the corrosion of copper, which was so little at the beginning of its use that it might be considered negligible, became important. Thus, in 1883, this corrosion was from five to six times as much as previously. Stolzel attributes that increase to the increase of the impurities in the copper The impurities in question were derived from the mineral matter in the foreign ore imported into England to meet the increased demand for the metal, and were in a very much greater proportion than in the purer native ore that had previously furnished the copper needed.

At this epoch Davy experimented on the protection of the sheathing-copper of wooden sea-going vessels by placing it in contact with electro-positive metals that were more easily attackable, such as iron and zinc. He ascertained that the protection was complete as long as the surface of these metals was maintained at from the $\frac{1}{40}$th to the $\frac{1}{150}$th of the surface of the copper to be protected. But, unfortunately, this process produced carbonates of lime and of magnesia which increased the development of barnacles and of algae, this causing a still greater increase in the resistance of the vessels. Nevertheless, further researches, confirmed by practice, showed that by adopting from the $\frac{1}{150}$th to the $\frac{1}{1000}$th for the ratio of the surfaces referred to, this disadvantage disappeared and the copper remained clean and free from attack. Ultimately, however, the method was definitively abandoned owing to practical reasons. Later, the investigation was again undertaken in the laboratory of the Toulon arsenal, by Becquerel, who attributed the practical failure of the method to the fact that the protecting metal disappeared much more rapidly than had originally been foreseen.

Since 1883, this question has again arisen, not for the protection of the external sheathing of wooden vessels, but for the preservation of the copper and brass tubes of the surface-condensers of marine steam-engines, and for the preservation of their brass feed-pumps and bilge-pumps, etc., the rapidity of the destruction of which in a large number of merchant ocean-steamers seemed so great as to have caused the abandonment of copper and its alloys as too costly in favor of the use of wrought and of cast iron, notwithstanding the inconvenience of the frequent renewals of the latter needed. The cause of this corrosion of copper and of its alloys, is certainly of an electrolytic nature, and should be attributed to the impurities in the metal, but the kind of these impurities and the forms under which they are present play a more important part than their proportion in the metal. The remark should be here made that copper refined in the dry way by the ordinary processes, but still containing a little oxide, is less attackable than the chemically pure copper obtained by electrolysis.

The alloys of copper have given no more satisfactory results than that metal itself. The demand of the specifications, for example, that the foreign matters should not exceed six-tenths of one per centum of the copper, has no great importance as regards resistance to the attack of sea-water.

Finally, the concentration, the composition, and the temperature, of the sea-water, seem to have great influence on the rapidity of the corrosion. Thus, a sea-water more concentrated with marine salt, richer in chloride of magnesium, and hotter, as in the case of the condensing water flowing from a surface condenser; or, simply in the case of tropical sea-water near certain coasts, is greatly the most corrosive.

All these facts show clearly that no great protection is to be had from any modification in the composition of the copper, and that the best chance is an investigation whether a coating or covering—metallic or non-metallic—could be found that would hermetically separate the sea-water from the metal; such, for instance, as tar, asphalt, caoutchouc, varnishes, lac-

quers, tin, zinc, and the American coating invented by Sabine, which is one of the best. Nevertheless, none of these coatings has given satisfactory results. All of them are too fragile, wear off too rapidly and scale off; further, most of them very easily crack. The presence of these cracks causes the metal exposed by them to be attacked with much more intensity than if it had no protection.

Lead should make a sufficiently good covering, but, unfortunately for its efficiency, its thickness would have to be at least 0.12 inch, which would increase the weight of the tubes, diminish their diameter, augment their resistance to the flow of the condensing-water, and, finally, prevent the entrance of the expanding tool in those of small diameter.

Mr. Uthemann has likewise had recourse to a covering, but of a different nature from the preceding ones; it is an oxide of iron deposited by a strong electrolytic action upon the copper immersed in a mass of sea-water. The idea of such a coating was suggested to him first by the experiments of Davy and of Becquerel, and then by the well known practical fact that the copper or brass tubes of surface condensers with cast iron shells are but slowly attacked by sea-water.

As an intense electrolytic action can be produced at only a short distance, the piece of copper or brass to be protected is enveloped by a wire or band of iron preferably to a band of zinc for the material, wound spirally around the piece; the zinc is, in fact, less electropositive than the iron, and, further, it has the inconvenience of dissolving in sea-water. If a copper-tube thus prepared be immersed in sea-water there will be a copious disengagement of gaseous bubbles and the liquid becomes simultaneously of a reddish-yellow color by reason of the formation of the oxide of iron at the expense of the iron.

At first, the copper remains bright, then the oxydation of the iron follows and the copper becomes covered between the spirals with iron which soon becomes sufficiently adherent not to be swept off by a very rapid current of water, even after the removal of the iron spirals. The same deposit is pro-

duced, but more slowly, under the spirals themselves. If the coating be scratched by a scraper or sharp edged tool, the copper beneath appears bright and unchanged. When all the surface of the copper is covered with a sufficiently thick coating, the attack of the iron seems to cease and the piece can be removed. The copper can then remain indefinitely in the sea-water without alteration, provided the coating be maintained in its integrity. The same results are obtained with the alloys of copper.

If the tubes, or other pieces, be formed of pieces of different metals soldered together, one piece being of copper and the other pieces being of brass or other alloys of copper, the corrosive action is the same. Coated tubes, that is to say, tubes coated with tin or zinc, resist corrosion better than non-coated tubes.

An experiment made with a copper tube coated with iron electrolytically deposited upon it showed that this coating of metallic iron dissolved entirely off in sea-water, and affords no protection because the surface of the copper becomes rapidly exposed. Only the oxide of iron produced by the process hereinbefore described is protective, and not the metallic iron itself. As the protection is the same whether given to copper or to its alloys, the employment of pure copper is preferable to the employment of the alloy of 97 per centum of copper and 3 per centum of zinc which has frequently been recommended as less attackable by sea-water, but it is more expensive and not as easy to work as copper.

This mode of protection is just as properly applicable to the piping of bilge-pumps, as the water pumped by them is generally acid and strongly corrosive; and to the channels of frigorific apparatus using a highly concentrated brine as the non-congealable liquid.

Mr. Uthemann has made a great number of experiments of the kind precedently described. He has operated on copper tubes subjected to both flowing water and to stagnant water. These experiments have extended over two years.

The arrangements adopted in the first case are represented

in Figure 1. A and B are the tubes operated on; they are placed in the circuit of an electrically driven pump which draws the water from a reservoir situated beneath them and delivers it into the same reservoir after its passage through the tubes.

The conditions under which these experiments were made, reproduce as closely as possible those of the copper-conduits

Fig. 1.

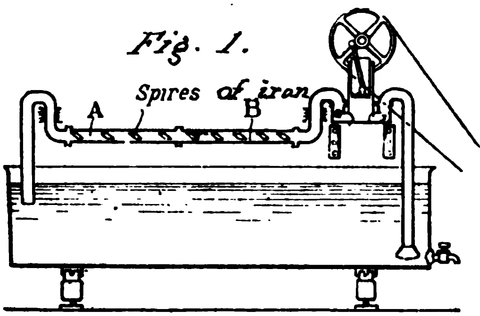
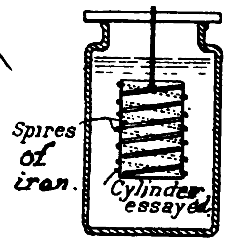


Fig. 2.



ARRANGEMENTS ADOPTED FOR THE EXPERIMENTS WITH FLOWING WATER AND WITH STAGNANT WATER.

of sea-going vessels. The tubes A and B, like them, are subjected to the double influence of air and of sea-water which are the most favorable to their destruction.

The apparatus was in use ten hours per day, and, owing to the arrangement of the tubes, the water remained in them during the rest of the day when the pump was not working.

The water was renewed every fortnight, and the temperature of the laboratory was maintained at 68 degrees Fahrenheit.

The accompanying table contains the results of these experiments.

Tubes of electrolytic copper and of ordinary copper were experimented with. These tubes were cleaned, polished and weighed, before each experiment. The experiments were simultaneously made, and under precisely the same conditions, on protected and on non-protected tubes, consequently, the results are rigorously comparable.

Also, the composition of the sea-water and the mode of assembling the tubes were varied; and, finally, for ascertaining whether the protection were localized or communicated to adjacent parts, an experiment was made in which only one of the two tubes was protected. After each experiment, the iron wire wound spirally around the tubes, which constituted the protection, was taken off, the tube cleaned and washed with water, alcohol, and benzine, and then weighed.

Any increase of weight (due to the deposition of the oxide of iron) indicated protection; any decrease of weight indicated lack of protection, and, therefore, an attack on the copper; this loss of weight increases with increase of time.

The experiments in stagnant water were made on cylinders of thin copper, arranged as shown in the marginal figure 2; that is to say, in a glass vase. Cylinders were tried both of brass and of copper, or made one-half of each of these metals.

The tubes, both of copper and of brass, used in surface-condensers, were also experimented with by this method. All these tubes were, like the others, polished and weighed before being experimented with. The spirals of protection were applied either solely on the interior, or, simultaneously, both on the interior and the exterior of the tubes. The temperature was 68 Fahrenheit degrees, and the sea-water, taken from the Gulf of Danzig, was vaporised until it contained 3½ per centum of salts. Mr. Uthemann employed a solution of 3 per centum of ordinary salt and 0.5 per centum of chloride of magnesium. Finally, for the purpose of ascertaining whether the protection of the oxide of iron was obtained in acidulated water, he added, also, in certain cases, 0.1 per centum of hydrochloric acid.

At the commencement of the attack of the iron, there was formed on the protector a great quantity of rust which necessitates rinsings of the tube before its use. This formation of rust deteriorates but little the protecting steel spirals; one of these spirals one-fiftieth of an inch in diameter, after a year's use had lost nothing of its cohesive strength or of its elasticity.

TABLE CONTAINING THE RESULTS OF THE EXPERIMENTS MADE BY MR. UTHEMANN, MEMBER OF ENGINEER COUNCIL OF THE GERMAN NAVY, TO DISCOVER A PROCESS WHICH WILL PREVENT THE CORROSION BY SEA-WATER OF COPPER AND BRASS UNDER THE CONDITIONS FOUND IN THE SURFACE CONDENSERS OF MARINE STEAM-ENGINES.
"Le Génie Civil" of 23d September, 1905.

Number of the Experiment.	Duration of the Experiments, in hours.	Metal essayed.	With or without protection.	Increase or diminution of weight, in fractions of a pound per square foot of surface of the metal essayed.	Conditions.
				1. In Flowing Water.	
1a	94	Cast copper	With	+0.017026322	Water containing 3.5 per centum of marine salts. Temperature 55.4 degrees Fahrenheit.
1b	94	do.	Without	+0.001980663	
2a	52	Cast copper	With	+0.020224908	Water containing 3.5 per centum of marine salts. Temperature 60.8 degrees Fahrenheit.
2b	52	do.	Without	+0.007922653	
3a	30	Electrolytic copper	Without	−0.005240751	Water containing 3.5 per centum of marine salts. Temperature 60.8 degrees Fahrenheit.
3b	30	do.	Without	−0.008527924	
4a	187	Electrolytic copper	With	+0.034480762	Water containing 3 per centum of marine salts, and 0.5 per centum of chloride of magnesium. Temperature 64.4 degrees Fahrenheit.
4b	187	do.	With	+0.041596387	
5a	187	Electrolytic copper	Without	−0.025721556	Water containing 3 per centum of marine salts, and 0.5 per centum of chloride of magnesium. Temperature 64 4 degrees Fahrenheit.
5b	187	do.	Without	−0.036089897	
				2. In Stagnant Water.	
1a	59	Electrolytic copper	With	−0.021715942	Sea-water containing 3.5 per centum of marine salts, and 0.8 per centum of hydrochloric acid. Temperature from +41 to −4 degrees Fahrenheit. Iron spirals on the outside only.
1b	59	Brass	With	−0.032163017	
1c	59	Commercial copper	Without	−0.180326464	
2a	33	Electrolytic copper	With	+0.003533208	Sea-water containing 3.5 per centum of marine salts, and 0.8 per centum of hydrochloric acid. Temperature from +41 to −4 degrees Fahrenheit. Iron spirals on the outside only.
2b	33	Brass	With	+0.006677664	
2c	33	Commercial copper	Without	−0.004345157	
3a	198	Electrolytic copper	With	+0.013163414	Sea-water containing 3.5 per centum of marine salts. Temperature 64.4 degrees Fahrenheit. Iron spirals on the outside only. The cylinders were covered before commencing the Experiment with a deposit of electrolytic Iron.
3b	198	do.	Without	−0.027232273	
3c	198	Brass	With	+0.005181710	
3d	198	Commercial copper	Without	−0.212829023	
4a	74	Cast copper	With	+0.001001404	Sea-water containing 3.5 per centum of marine salts. Temperature 71.6 degrees Fahrenheit. Iron spirals on the outside only.
4b	74	do.	Without	−0.027360217	
4c	74	Brass	With	+0.000058559	
4d	74	Commercial copper	Without	−0.020766208	
5a	74	Condenser tube of Heckman bronze	With	+0.001933915	Sea-water containing 3.5 per centum of marine salts. Temperature 71 6 degrees Fahrenheit. Iron spirals on both the outside and the inside.
5b	74	Brass	With	+0.000880841	
5c	74	Heckman Bronze	Without	−0.024456884	
5d	74	Brass	Without	−0.014762707	

NOTE BY THE TRANSLATOR.

The process set forth in the foregoing translation, is of a wholly new and very interesting method of preventing the corrosion by sea-water of copper, brass, and bronze under the conditions as nearly as they could be reproduced in which these metals are employed for the surface liquefaction of the exhaust-steam in the condensers of marine steam-engines. As all the other processes which have been tried for this purpose have been failures more or less signal, an entirely new one, supported by extensive experimental results, has great interest in steam-engineering, and even if it should not prove to be practicable, itself, on a commercial scale, it may be valuable as a stepping-stone to something that will be commercially successful in a very important engineering subject. The process in question, and its results under the experimental conditions, of course, cannot be accepted as a finality until tested for a considerable length of time on the large scale and under the conditions of actual practice. But, under any circumstances, the facts developed are suggestive to all physicists and increase largely the sum of this kind of knowledge.

The general facts regarding the corrosion by sea-water of copper and brass, are substantially the same in the case of the sheathing by these metals of wooden sea-going vessels, and in the case of the copper and brass tubes employed in the surface-condensers of marine steam-engines.

In both cases, the corrosion in question is of two kinds, one a uniform oxidation of the metal without sensible electrical phenomena, commencing with the first exposure and continuing until the final disappearance of the metal, the oxide and other products chemically produced having scarcely any adherence to the metallic base on which they are formed, and on which they merely rest as a sort of dust that continually washes off when the metal is laved by water, the washing off being so much the more rapid as the velocity of the washing water is greater. This continual oxidation and washing off of the oxide, constitutes the peculiar fitness of copper and brass sheathing for sea-going wooden vessels as a preventive

of the accumulation of marine animals on their immersed surfaces, a growth that not only greatly increases their resistance to movement in water, but, in the seas infested with the "teredo-navalis," prevents the destruction of the wood itself by these ship-worms. In the case of the metallic sheathing, however, the continual washing off of the non-adherent oxide carries with it this sea-life as fast as formed upon it, and leaves the metal clean, thus avoiding the increased resistance due to such an accumulation. The sea-life develops as rapidly upon the oxide as it would upon the unprotected wood. There is no poisonous effect produced on the sea-life by the metal or by its oxide, the barnacles simply fall off as the non-adherent oxide to which they clung falls off. Neither does the smoothness of the metallic sheathing prevent the attachment of the sea-life to it, as that attachment takes place to glass as rapidly and as firmly as to wood. The entire beneficial effect is due merely to the fact of the non-adhesion of the oxide of copper and brass to its metallic base, being in this particular the very opposite of the strongly adherent oxide of iron to its metallic base.

Sea-water is a solution of strongly alkaline salts, and its corroding effect upon copper and brass is in some ratio to the percentage of these salts in it. This effect is increased by increased temperature of the solution and by the rapidity of the movement of the metals in it, and is decreased in some ratio to the mass of the metal exposed, all other things remaining the same. As the mass of the exposed metal is increased, not only does the *percentage* of the weight of it corroded in equal time decrease, but the *absolute* weight of it corroded in equal time decreases also. This is an effect due wholly to mass, the greater mass having greater resisting power to chemical reaction. A neglect of this fact in many experiments on this subject, have caused erroneous *quantitative* conclusions, although the results may hold true *qualitatively*. The specimens of different metals experimented with should have the same weight, and, as nearly as this premiss will allow, the same dimensions.

The copper and brass sheathing of wooden vessels corrodes faster in sea-water when the vessels are in motion than when they are motionless, because, in the latter case, there is none of the washing action that exists in the former case and removes the oxide as fast as it is formed. When the vessels are motionless, the oxide adheres longer to its metallic base, and thus, in measure, reduces the rapidity of the corrosion by acting as a covering to the metals. The same result due to the same cause occurs with the copper and brass tubes of the surface condensers of marine steam-engines. But here must be remarked the important fact that as vessels not under way generally lie at the wharves of cities, and that as the sea-water in which they float is always more or less much polluted by the sewage discharged into it, the rapidity of the corrosion is increased, and often in a high degree by this pollution, so that in comparing the amount of corrosion produced in equal time with the vessel in motion and at rest, allowance must be made for the fact that the sea-water in the two cases is generally not of the same composition. Further, even were the sea-water of the same composition, there still remains the disturbing fact that the vessel at the wharf is subject to the movement of the tides and other currents over its sheathed surface which prevent a strictly quantitative comparison of the corrosion results in the two cases of motion and at rest. In this respect the conditions for the sheathing varies from those for the tubes in the condenser; the corrosive solution does not move over them when the vessel is at rest, and ought not to be present, but, as owing to the leakage of sea-valves, the condenser in such case is generally filled with outside water up to the outside level, and which remains in them practically motionless, the corrosion of the tubes will be less when the vessel is at the wharf than when it is in free route, the composition of the sea-water being the same, nevertheless, corrosion even then will be more or less in progress.

The corrosion hereinbefore described, is, as was stated, general in character and uniform in effect; it was a coating removed from the entire surface of the metal, leaving the

remainder clean and smooth. This corrosion was comparatively slow in its progress, and the injury done by it was not commercially important. Of course, it involved, after a time, a replacement of the metal, but the time at which this had to be done could be long foreseen, so that the inconvenience was comparatively small.

But there is another kind of corrosion in progress nearly simultaneous with the corrosion just described, and it is greatly more destructive and inconvenient. It is local instead of general, and attacks the metal in the most capricious manner, leaving some portions untouched while adjacent portions are absolutely destroyed to dust and gone. This is the kind of obvious corrosion which is attributed to want of homogeneity in the metal, and to electrical action, when the metal is in contact with sea-water. It commences by the appearance of extremely small holes—pinholes—more or less near each other, but only in particular spots on the metal, which holes gradually enlarge until they become sometimes confluent like the pustules of the smallpox, and sometimes they permanently preserve their circular form, the corrosion ceasing at that place and not extending farther, the edges or sides of these holes being nearly at right angles to the surface of the metal, and showing nearly the original thickness of the metal. They appear like holes artificially bored through. In this manner half of a tube in a surface-condenser will sometimes be completely destroyed while the remaining half will show no appearance of attack. Again, this kind of corrosion will continue slowly until it extends throughout the entire tube, which, while preserving its general form, has lost weight, color, and cohesive strength, and crumbles into dust by handling. Given, however, sufficient time, the copper or brass tube will disappear, having been apparently wholly dissolved by the sea-water. Or, more properly, the metal having become wholly chemically combined with the marine salts, and with the gases dissolved in the water, the results of the combination being various products in the liquid state, none of the metal being recoverable, as such, by evaporation. The pro-

cesses of the various reactions are exceedingly complex and obscure, and have never been convincingly investigated. All that is known is the final effect as shown by practice on the large scale and extending over, comparatively, considerable periods of time. Laboratory experiments are indeed most valuable, but only as guides to trials under the actual conditions of practice. As regards either palliatives for or preventions of, the corrosion in question, the commercial success involves not only the chemical success but also the money cost of the system; if that cost be greater than the cost of replacing the deteriorated metal, the method will be practically a failure however great its scientific success.

The same processes, appearances, and results, characterize the corrosion of the metal when employed as an external sheathing to the immersed part of the hull of a sea-going wooden vessel, as in the case of the tubes of the surface-condenser of a marine steam-engine.

What has hereinbefore been stated, refers to the corrosive action of pure or normal sea-water, but when this water is mixed with putrescent vegetable matter, reactions take place between them which convert the sea-water into what is known as "bilge-water," the corrosive power of which on copper and brass is much greater than that of normal sea-water, so that the destructive action of the latter upon these metals is very considerably accelerated.

The term "bilge-water" shows that this water was first observed in the "bilges" of wooden sea-going vessels, the putrescent vegetable matter required to be mixed with normal sea-water to form it, being supplied in that case by the decaying wood of the vessel, while the sea-water was supplied by inleakage from without. Bilge-water will rapidly destroy copper, brass, and iron pipes, made of *thin* rolled metal exposed to it; but cast copper, cast brass, and cast iron pipes, being made much *thicker* in material, resist much longer under the same circumstances than in the ratio of the difference of the thicknesses of the rolled and cast material.

Again, bilge-water is found on a large scale at the wharves

of cities laved by ocean tides, and whether these wharves be built of wood or stone; the putrefying vegetable matter being abundantly furnished by the sewerage discharge of these cities into the sea-water at their wharves. Hence, the rapid destruction that has been noticed in many cases of the copper and brass sheathing of wooden vessels lying for any considerable time at such wharves. Moreover, the iron boilers of steam-engines fed with this water are found to be not only rapidly corroded out, but kept free of sea-water scale by using this wharf-water; and the same effect is produced by feeding the boilers of marine steam-engines having jet condensers on board of wooden sea-going vessels with bilge-water instead of with normal sea-water from the hot-well.

Bilge-water is the best of all removers of sea-water scale from boilers, but it will, at the same time, attack their material.

In the chemical reactions between putrescent vegetable matter and the various dissolved salts of normal sea-water, the sulphur salts of the latter appear to furnish by their decomposition the sulphur which combines anew with the decaying woody matter. These new sulphur-compounds seem to be the corrodents carried by the bilge-water as a vehicle, and they are as corrosive in their vaporous as in their liquid state. They vaporise with the water, and have the strong and nauseating stink which characterises bilge-water; so much sulphur, so much stink. This vapor also reacts energetically on silver, on iron, on white-lead paint, and on a great many other substances.

The fetid odor of the gases continuously discharged from bilge-water compared with the nearly odorless steam evaporated from normal sea-water, and the strongly corrosive reaction of those gases compared with the neutral effect of steam from normal sea-water, show how great are the chemical transformations produced by the addition of putrid vegetable matter to sea-water. The transformations in question occur at all temperatures above the freezing point of the water, but they take place the more rapidly the higher the temperature.

Healthy wood, without rot and entirely submerged in sea-water, does not undergo decay as long as it is kept submerged. In that condition it will remain unaffected indefinitely, but it must not be alternately submerged and exposed to the atmosphere at intervals; the immunity only exists under the condition of uninterrupted exclusion of the atmospheric gases; as soon as they obtain access to it decay commences.

The *Chicora*, a wooden sailing merchant vessel employed in the China trade, was newly sheathed in January, 1847, with 7,392 pounds of copper, which was corroded so rapidly that the vessel had to be resheathed in March, 1849, only 2,682 pounds of the metal remaining. Time, 26 months.

The *Hamilton*, also a sailing wooden merchant vessel employed in the India trade, was newly sheathed in October, 1847, with 7,706 pounds of annealed copper, which corroded so rapidly that it had to be removed in August, 1849, only 3,086 pounds of the metal remaining. Time, 22 months.

The *Carthage*, a wooden sailing merchant vessel, also employed in the India trade, was newly sheathed in November, 1847, with 8,727 pounds of cold-rolled copper, which was so rapidly corroded that only 5,810 pounds remained when the vessel was resheathed in August, 1849. Time, 21 months.

The average duration of copper sheathing on American wooden vessels engaged in foreign trade, was 3 years, at the expiration of which time what remained of it had to be removed. The sheathing was of ordinary commercial copper.

The wooden vessels of the United States Navy, were formerly sheathed with copper refined to the highest degree in the Washington Navy Yard where the Government maintained a special plant with which no expense was spared in purifying the sheet-copper. The average duration of this sheathing on these vessels was about 5 years.

A number of years ago a wooden vessel belonging to the British Navy, and engaged in the suppression of the Slave Trade on the western coast of tropical Africa, had its copper-sheathing destroyed in an unprecedentedly short time. This vessel lay closely off a low coast of swamps of immense extent,

the tide covering and uncovering them alternately; the decaying matter of the swamp vegetation becoming largely mixed with the surface sea-water, formed doubtless the resulting principal corrosive agent, namely, sulphuretted hydrogen.

Similarly, the land above the Mare Island Navy Yard in California, is covered to an enormous extent with fresh-water swamps, the water of which, impregnated with the putrid parts of this continually dying vegetation, flows slowly past the wharves of the Navy Yard where it is mingled with the mixture of sea-water and fresh water from the adjacent bay and the rivers emptying into the bay. This mixture has an average density of five-eighths of the density of natural sea-water, and is so corrosive that the iron boilers of the shops of the Navy Yard using it for feed never form scale; their inner surfaces are always clean and undergo rapid destruction by corrosion.

The feed-water pipes and the bilge-water pipes of the United States wooden steam-frigate *San Jacinto* were wholly destroyed by corrosion in about 5 years. These pipes were of copper, and being situated in the bilge of the vessel were exposed to the action of the bilge-water which was the corroding agent.

During the same time (about 5 years), the tinned brass tubes of the Pirsson surface condenser of the engines of the *San Jacinto* were so much corroded by the sea-water passed over them for the liquefaction of the exhaust steam that they had to be removed. The shell of the condenser was of cast-iron, and but slightly affected.

The United States screw-steamer *Princeton* was constructed of green white oak. The bilge-water in this vessel was extremely fetid, so much so that the engine-room personnel had to hold their noses when they first went on watch until they became gradually accustomed to the stench. When the vessel was under steam there was a copious admission of sea-water to the bilges at the stern stuffing-box of the screw-shaft. The engines, two in number, had jet-condensers, and the boilers were fed with sea-water from their hot-wells. The *Prince-*

ton had no bilge-pumps worked by the engines, but the bilge-water was pumped out by the vertical single-acting air-pump of one of them, which pump was fitted with a snifting-valve at its bottom; whenever this valve was submerged by the bilge-water, it was opened by hand and held open until the bilge was cleared of water. Now the bilge-water, by this arrangement, was pumped into the hot-well and thence overboard through the outward delivery-valve, a part of it passing from the hot-well with the normal hot-well water into the boilers as a portion of their feed. The bilge was pumped out about every two hours. The density of the water in the boilers was maintained at twice the density of sea-water, and the steam-pressure averaged about 12 pounds per square inch above the atmosphere. Yet no scale ever formed on the interior surfaces of the boilers, but their metal corroded with extraordinary rapidity. In about 4 years' use they were corroded beyond repair, although, during that time, they were once extensively repaired. In about the same time (4 years' use) the brass of the air-pump, the feed-pump, and of the outboard delivery valve, and the cast-iron of which the connecting passages was made, had become so corroded as to need renewal, while the copper feed-pipe, which was situated low down in the bilge, was continually under repair. Now the corresponding parts of the other engine, which had been subjected to the action of only the normal sea-water of its air-pump discharge, remained unaffected, showing that the corrosion in question was due wholly to the bilge-water pumped overboard by the engine discharging it.

The corrosive effect of sea-water on metal is much intensified by mixing it, or supersaturating it, with atmospheric oxygen as in the case of spray. A striking example of this was shown at Vera Cruz in Mexico during our war with that country. Two medium sized cast-iron cannon had been placed vertically at the foot of the Mole or long pier extending into the sea; they were sunk, to act as posts, into the earth for about one-third of their length, and during strong winds were exposed to the dash of fine spray driven against

them from the sea, and always in the same direction; the result was that the side of the cannon receiving the spray had been so much corroded that about one-third of the material had disappeared, the cannon having a flat surface on that side instead of a round surface, while the opposite side was not affected. The cannon were about 50 feet apart, and both were corroded in the same manner and to the same extent. It is quite possible that the sea-water pumped as injection through the tubes of the surface-condenser of a marine steam-engine, and supersaturated with atmospheric oxygen due to the dashing of the waves into the air, caused by movement of the vessel, or by winds, may, in this manner have its corrosive power considerably increased. In other words, the corrosive effect of sea-water supersaturated with oxygen, is probably much greater than when not thus supersaturated.

There is still in the case of the surface-condensers of marine steam-engines, another possible cause increasing the corrosive effect of normal sea-water, namely, the electricity developed by the friction of this water upon itself and upon the metallic surfaces on which it flows.

However small these causes may seem to be, *per se*, yet when the relatively trifling weight of the metal subjected to the corrosion is considered in comparison with the very large weight of water coming in contact with it during considerable intervals of time, even a small increase of the corrosive power of the water becomes important commercially.

Some years ago, a large United States wooden steamship was fitted out at the Brooklyn Navy Yard for an extended cruise. The vessel which had just been newly coppered, lay in tide water at the Navy Yard Dock, and opposite to the discharge of one of the large sewers of the City of Brooklyn. The result was that when the vessel arrived in direct transit at Rio de Janiero in Brazil, the copper sheathing was so completely corroded that it had to be entirely renewed; it was probably greatly disintegrated before the vessel left the dock. The mixture of the putrefying vegetable matter of the sewage with the sea-water had formed a liquid (real bilge-water) that

dissolved the copper. The stench of the sulphuretted-hydrogen gas emitted by this liquid was nauseating in the extreme. The dock was a wooden platform resting on wooden piles partly out of water and partly in water. The tidal sea-water flowed among and around these piles.

The following are the generalisations which may be considered as warranted by the observations made on the large scale under the conditions of practice, regarding the corrosion of copper, brass, and bronze, in normal sea-water, and in sea-water contaminated by putrefying vegetable matter (bilge-water).

(*a.*) Normal sea-water contains dissolved in it sulphur compounds. Putrefying vegetable matter is dead vegetable matter, the constituents of which are continuously undergoing changed relations of chemical affinity, the affinities that existed between them in the living vegetable organism becoming continuously lessened to complete extinction in the dead vegatable matter, so that new chemical combinations can easily take place. Especially in this the case as regards the hydrogen-compounds is vegetable matter; the force of the chemical affinities of the constituents of these compounds greatly lessens in the change of the vegetable matter from life to death.

When, therefore, the dissolved sulphur-compounds in the sea-water, are presented to the hydrogen-compounds in the changing dead vegetable matter, there occurs a double decomposition, and a new compound, sulphuretted hydrogen, is formed; sometimes, also, there can be formed, additionally, hydrochloric acid by the combination of the hydrogen freed in the nascent state in the decaying vegetable matter with the chlorine of the dissolved common salt in the sea-water. Now, both the sulphuretted hydrogen and the hydrochloric acid have excessive corrosive action on metals. Thus may be easily seen that, according to circumstances, according to the more or less decay of the vegetable matter, the corrodents so generated in equal time in a given mass of sea-water, may be greater or less in quantity and in virulence of action;

and that the corrosive action due to the alkaline salts dissolved in normal sea-water may be greatly added to by the new sulphur compound, and by other compounds, generated as above described, and which convert normal sea-water into bilge-water, the latter being by no means confined to the bilges of wooden ships, but being found wherever sea-water is mingled with putrefying vegetable matter.

For normal sea-water, the corrosive action is due purely to the alkaline salts dissolved in it and which undergo decomposition in the presence of the metal, and destroy it by forming new compounds with it, thus forming one kind of corrosive action; while, for bilge-water, another kind of corrosive action is added, thus accounting for the variety in the corrosive phenomena, and for the difference in the rapidity of the action. There may, also, be stated that nearly all vegetable matter contains a little sulphur in some form dissolved in its sap and incorporated as foreign matter in its fiber.

The hydrogen and oxygen of cellulose, when the cellulose decays and disintegrates, may decompose from their combinations and the former may unite with the sulphur of the sulphur salts dissolved in sea-water, and, in some cases, with the chlorine of the common salt dissolved in sea-water. Further, the oxygen constituent of the cellulose, liberated in the nascent state furnishes an extremely efficient corrodent. Some electrical action no doubt is likewise a factor.

The whole subject is very obscure and complex, but not so confused that a tolerably accurate idea cannot be formed of the causes and consequences of the corrosion in question.

(*a.*) The corrosion of the metal is in all cases more or less rapid as it is more or less impure, not only because the impurities may be more easily acted upon, but also because by separating the molecules of the pure metal farther apart, its cohesive strength is thereby correspondingly impaired.

(*b.*) As regards the temperature at which the corrosion takes place, the higher the temperature of the metal, the more rapid is the rate of the corrosion. This result is doubtless due to the fact that the density of the metal decreases as its

temperature increases, its molecules being kept at a greater distance apart with higher than with lower temperatures whereby the force of the intermolecular or cohesion attraction of the metal is lessened in some ratio to its increase of temperature, and its resistance to chemical combination with other substances is thus correspondingly lessened.

(*c.*) The larger the mass of the metal exposed to corrosion, other things equal, the smaller will be the rate of its corrosion; that is to say, a less absolute weight of metal will be destroyed the greater its mass.

In the case of condenser tubes, or of the metal sheathing of wooden ships, the thicker the metal the slower will be the rate of its corrosion, and not only relatively but absolutely. As regards the condenser-tubes, the thickening of their metal involves not only increased money cost at first, but increased weight to be permanently carried afterwards, and somewhat increased dimensions of the condenser. This money loss at first, however, is recouped at last by the greater durability of the tubes above the proportionality of their increased weight, leaving out of consideration the valuable convenience of their longer endurance which is a merit of importance especially for naval steamers. The thickness now used for the metal of the tubes should be doubled.

The gain of resistance to corrosion in the case of increased thickness of metal, seems to come from the fact that the action of the corroding power is limited to the surface exposed, while the resistance to it is, in some ratio, proportional to the mass. The combined resistance of the intermolecular attraction of the mass seems to be a factor in lessening the corrosion in a higher degree than the increase of the mass.

(*d.*) Coating the tubes with tin delays their corrosion as long as the tin lasts, but its durability is not great, and its very slight thickness on the tubes cannot be increased beyond what is obtained from the first "dip." This very small thickness of tin prevents the action of the tube-metal from taking an increased thickness of tin from additional "dips;" and it also diminishes to an appreciable degree the heat conducti-

bility of the tube; nevertheless, this "tinning" should never be omitted, although it involvės additional surface of tubes for equal heat-conduction in equal time.

(*e.*) In surface-condensers the temperature of the exhaust-steam is always higher than the temperature of the metal of the tubes in contact with it; and the temperature of the metal of the tubes is always higher than the temperature of the water of liquefaction in contact with it. This water, which is the normal feed-water, has, therefore, in the condenser in a marked degree, a temperature inferior to the temperature of the steam from which it is precipitated. Statically, such results would be impossible, but the case is one of dynamics, the heatings, coolings, and replacements being successive and time not being sufficient for obtaining equilibrium.

(*f.*) The air-pump draws from the surface-condenser not only the feed-water but also the unliquefied, exhaust-steam; it draws them simultaneously, and delivers them mixed together into the hot-well, consequently there occurs in this operation an impartation of heat by the exhaust-steam to the feed-water, so that the temperature of the latter in the hot-well may be higher than in the condenser. Furthermore, the air-pump compresses the exhaust-steam from its pressure in the condenser to at least the atmospheric pressure in the hot-well, if the hot-well be open to the atmosphere, thereby increasing the heat in that steam by adding to it the heat equivalent of the work of the compression.

(*g.*) The corrosion of the condenser-tubes is principally on their sea-water side, but their distilled-water side also suffers much, the corroding agents being the atmospheric gases mingled with the steam. This corrosion is quite uniform compared with the corrosion on the sea-water side, and the resulting oxides and salts are often carried into the boiler with the feed-water, and produce injurious effects upon the iron.

(*h.*) The more rapid the current of the sea-water over the surface of the metal, the more rapid will be the corrosion, as the oxides and salts of the corrosion having but slight adher-

ence to their metallic base, will be more speedily and thoroughly washed off, leaving the raw metal to be acted on.

(*i.*) In the case of horizontal condenser tubes, there will often be found a partial coating on their upper sides of baked mud, sometimes as hard as brick. This is mud deposited by gravity from the sea-water on the upper outer side of the tubes when the sea-water surrounds the tubes, and on the inner lower side when the sea-water passes through the tubes, and has been baked by the long application of the low temperature of the tubes. It is often mistaken for sea-water scale, but its nature may with certainty be known by mere inspection from the fact that it is not found on the lower outer side, or on the upper inner side of horizontal tubes, according as the sea-water is without or within them, nor on the outer or inner sides of vertical tubes. It is a great intercepter of heat and reduces correspondingly the steam-liquefying power of the tubes.

(*j.*) A surface-condenser should be opened at intervals, say, of six months; its tubes drawn out, cleaned, and examined with the particular view of ascertaining if any holes have been corroded through them by means of which the sea-water could find entrance into the distilled-water space. If any sea-water, however small in quantity, thus mingles with the distilled-water feed, its mineral matters held in solution will be precipitated upon the water-heating surfaces of the boiler in the form of scale. Such leakage will be shown by the "salinometer" if one be used, and one should always be in use, as it will indicate the gradual salting up of the water in the boiler even if the latter leak so that no undue accumulation of water in the boiler be apparent. Now, as there is always deficit of distilled water in all boilers under all circumstances, which deficit is restored by the "evaporator," a certain amount of leakage of sea-water into the distilled-water space of the condenser may not be shown by a rising water-level in the boiler, but will always be shown by the increasing density of the water in the boiler made evident by the "salinometer;" it will also be shown by the lessened quantity of distilled

water required to be furnished by the "evaporator." The presence of even a small proportion of sea-water in the water in the boiler can be discovered by a delicate taste, and also by any of the usual chemical tests. But the fact that the contamination of the distilled water in the boiler by inleaked sea-water is due to the perforation of the tubes in the condenser by corrosion, can only be shown by ocular demonstration.

THE PRESENT STATUS OF MARINE GAS ENGINEERING.

By Peter Eyermann, Associate.

[Continued from Vol. XVII, No. 2, "The Future of Marine Gas Engines."]

In presenting this second paper on Marine Gas Engines to the Society, I shall deal with the internal-combustion engine as it appears today in all kinds of services on salt water as well as on fresh-water vessels.

In my first paper I treated more particularly of the *future* development of the gas engine, and referred to the theory of the gas producers. In this paper I shall have to consider the *present* status and some details in regard to gas plants running on liquid fuels as well as on solid fuels, and also propellers and engine connections. Some recent lectures, as they appeared in Europe, and parts of which are translated, gave some interesting additions to this paper.

The development which the gas engine has undergone in the past ten years, especially in regard to reliability and the fuel-saving question, has influenced and encouraged both the builders and owners of vessels, and I think it worth while to present it in this paper. The first engines running in boats have been using illuminating gas. This gas was stored in big tanks under high pressure, as it was impossible to install an illuminating-gas factory on board. But the tanks required such a large amount of room and weight that they seriously influenced the tonnage of the vessel. This is the main reason why only a very limited number of boats driven by city-gas engines have been constructed.

In Paris there was a freight boat, *Idée*, running in the year 1895 between Havre and Rouen. The engine was of the two-cylinder vertical type, and is shown in Fig. 1. It had 10 H.P., and the fuel, city gas, was stored in 80 tanks, with

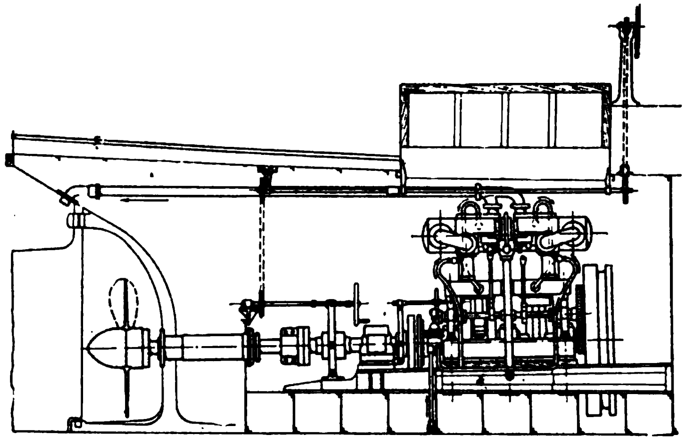


Fig. 1.

1,500 pounds pressure per square inch. Each tank had about 800 cubic feet capacity; therefore there were on board about 64,000 cubic feet. A good modern gas engine consumes about 16 cubic feet per H.P. per hour; thus, the steaming time of this ship was 100 hours at 6 miles per hour, and gave a radius of action of 600 miles. Each tank weighed 715 pounds, and the vessel was therefore carrying a dead weight of 57,200 pounds.

A good kerosene engine will require about one pound of fuel per H.P. per hour, or 40 pounds for a 40-H.P. engine. It will therefore be seen that this boat, when using kerosene, will have an action radius of 7,200 miles, if we assume the weight of the fuel tanks at only 4½ tons, which is a fair estimate.

It was not surprising that the illuminating-gas engine was soon displaced by the developing of the oil engines. The latter were independent of city-gas plants and compressing stations, and much cheaper in service.

The gasoline engine in the United States, and the kerosene engine in Europe, have been greatly developed since, and other liquid fuels, as alcohol, crude oil, etc., are also helping to supplant the steam engine.

One hundred and fifty years of marine steam engineering give, of course, some very valuable information as to "how to build a good marine gas engine." No reliable concern of today has had good luck in placing a stationary steam engine in a boat, or in using a marine steam engine for a stationary electric-power plant. It is useless to try this in gas engines, where the question becomes more complicated, from the simple rule, that each gas plant should be built to suit the specific fuel, and not a specific engine type suitable for all fuels. More and more are direct reversible marine gas engines being placed on the market, and the devices in use now, such as reversing coupling or reversing propeller, will disappear in the larger units.

A very light engine with a very high number of revolutions is the explosion engine for light boats and launches, and this is the marine engine of the present. Reliable, heavy engines, with thousands of horsepower, have not yet been built. Boat motors of today have 20 to 100 pounds per H.P., and stationary reliable engines have from 200 to 500 pounds per H.P. and more.

As the usual four-cycle, single-acting engine with only one cylinder is not at all equal, in regard to steadiness and vibrations, to the steam engine, two, three and up to six and eight cylinders have been installed, thus obtaining a motor which works very steadily, and is satisfactory as regards the balancing of the moving parts.

Experiments in this regard, as using two pistons in one cylinder, have been made, but proved to be not successful and too complicated. The cylinders of these have all been horizontal. As the greater number of gas engines today are non-reversing, I will first briefly touch on the required details of reversing propeller and reversing coupling.

It may be noted that in Europe the reversing propeller is just now more in favor than the reversing shaft coupling. Such a coupling is shown in Fig. 2; (a) is the rear crank of the gas-engine shaft, (b) the flywheel, (c) the propeller wheel, (d) the thrust bearing, and (e) the reversing clutch. A stronger

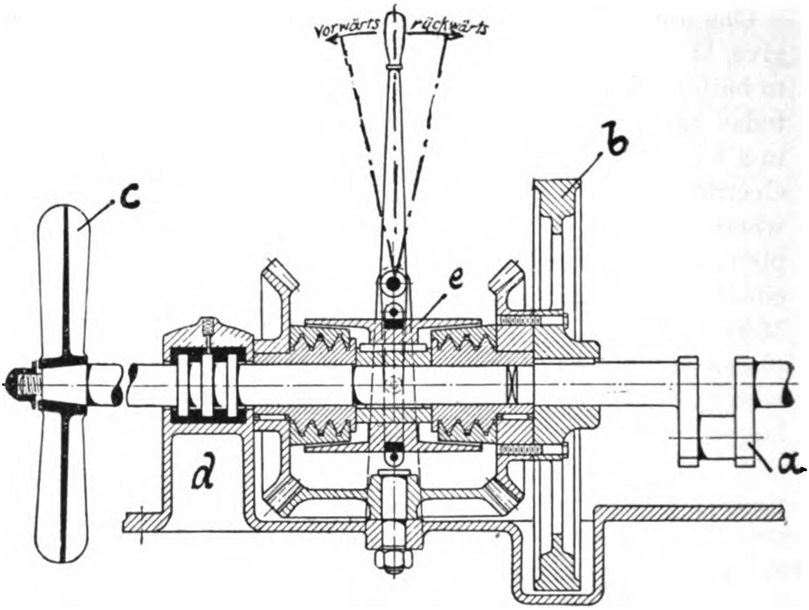
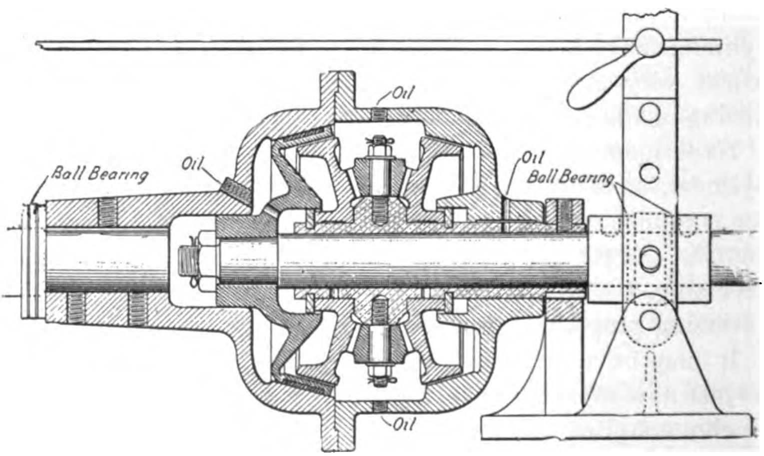


Fig. 2.

reversing coupling is shown in Fig. 3. The thrust of the propeller is directly on the friction cone, and sudden shocks on reversing are nearly avoided. All parts are enclosed in



REVERSING GEAR.

Fig. 3.

oiltight casings, and it takes up a very small amount of room. The weight of it, for a 2¾ shaft, is only 725 pounds.

In the last few years another type of reversing coupling has been developed—the so-called coil-spring friction clutch. These clutches are in use for 2,000-H.P. rolling-mill engines, and they will work well in boats. A picture for a boat outfit was shown in "Iron Age," December 14, 1905, and the arrangement for a 800-H.P. steam engine is as follows: The engine has about a 10-inch shaft and 80 revolutions. This speed is cut down to 40 revolutions, and the gear works on a 15-inch shaft. It is interesting from the marine engineer's view to know that the engine has a 46-ton flywheel near this clutch. The coupled continuation of the engine shaft has a spur gear with 27 teeth, meshing with a 54-teeth gear on the clutch shaft, and a 28-teeth gear on the back shaft; on the other end of the back shaft meshes a 45-teeth gear with a 90-teeth gear on the clutch shaft. All of these are spur gears and have a pitch of about five inches. It will be noticed that the two spur gears on the clutch shaft are arranged to run at about 40 revolutions, in opposite directions. By means of one or the other of the two coil clutches either of these may be engaged at will with the shaft. The thicker end of each coil has a lug formed on it resting in a pocket at the side of one of the gears, while the other end is connected to the short arm of a bell crank on the same gear. When the flanged collar is moved it comes into contact with the bell-crank lever of the corresponding coil, and the last turn of the coil is tightened on its drum, so that the entire clutch grips with great power

A good reversing coupling should have, however, the advantages of all spur gears, and that it occupy a relatively small space. In backing the speed is less, and the work for the gas engine easier. It is usually not desirable to run the propeller for backing as fast as for ahead, on account of the higher friction. This fact is sometimes the cause for sudden shutdowns of the engine, and is especially bad in smaller craft, where an easy handling is often necessary. It is worth while to note here that the usual gas engine of today reduces

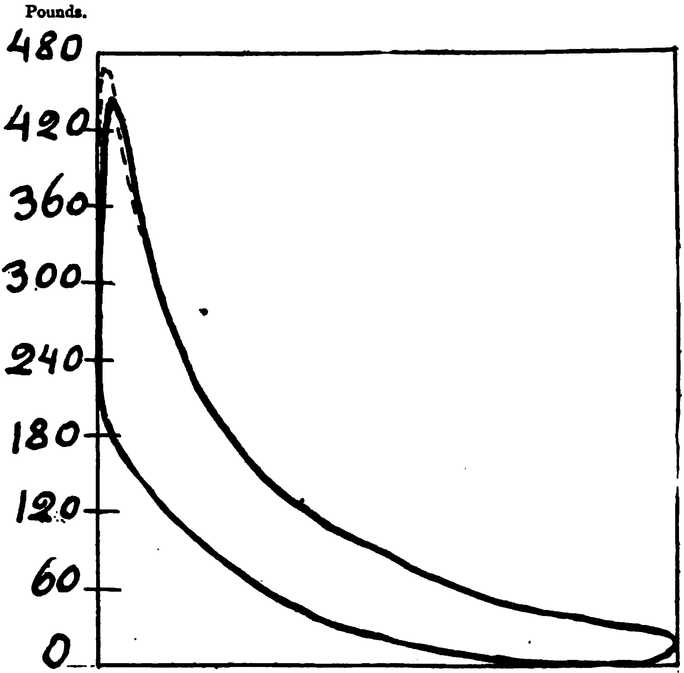
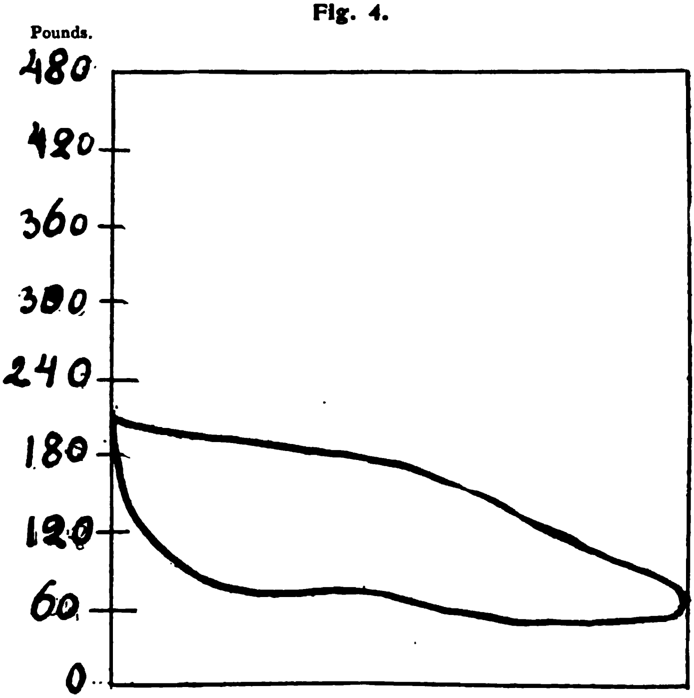

Pounds.
480
420
360
300
240
180
120
60
0

Fig. 4.


Pounds.
480
420
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240
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120
60
0

Fig. 5.

its energy and power with the reduction of revolutions, thus acting just contrary to the steam engine, where, with larger openings of the valve gear, even more power is obtainable under fewer revolutions. The slowing down of the vessel requires less revolutions, and under these conditions the gas engine works very uneconomically, as the exploded mixture has too much time for the internal losses, due to the cooling influence of the water.

Fig. 4 shows the diagram of a high-compression gas engine, and Fig. 5 the one of a high-pressure cylinder of a marine steam engine. The steam diagram may have about 90 pounds mean pressure and this gas engine diagram rather less, in spite of a six-times higher initial pressure for the gas engine. The three-throw crank shaft has about 14 inches diameter for a 2,000 H.P. steam engine; this would be, however, just sufficient for a gas engine having about half the power, namely, 1,000 H.P. or 1,200 H.P.

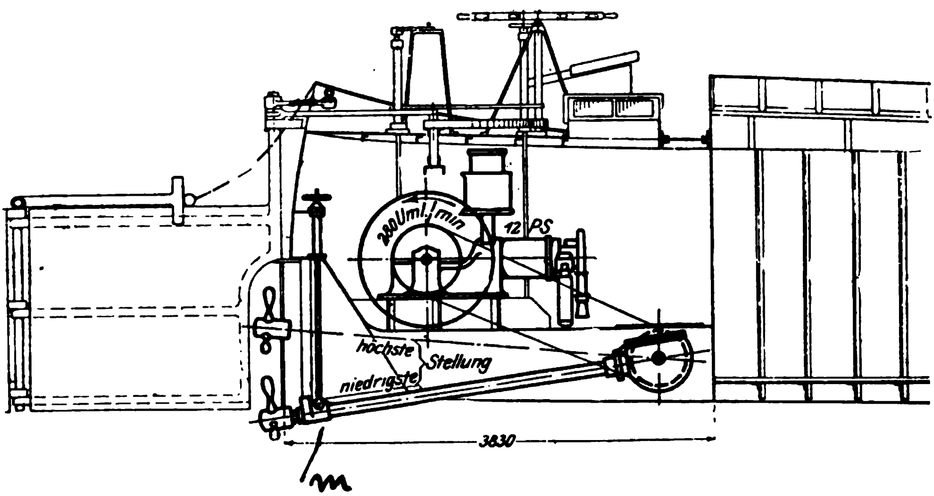


Fig. 6.

The second way of reversing was found in constructing reversible propeller wheels in connection with the non-reversible gas engine. A very interesting solution of a reversible propeller is shown in Fig. 6 and Fig. 7. The condition was to install the gas engine in a boat having 12 inches draught

at no load and 72 inches at full load, with a speed of about 1½ miles per hour. The latter was allowing a small propeller wheel, but the great difference in depth, at first, caused considerable difficulty. A solution was found in combining a liftable propeller wheel with a reversing mechanism. The force is transmitted by the shaft (a), through the rotating gear

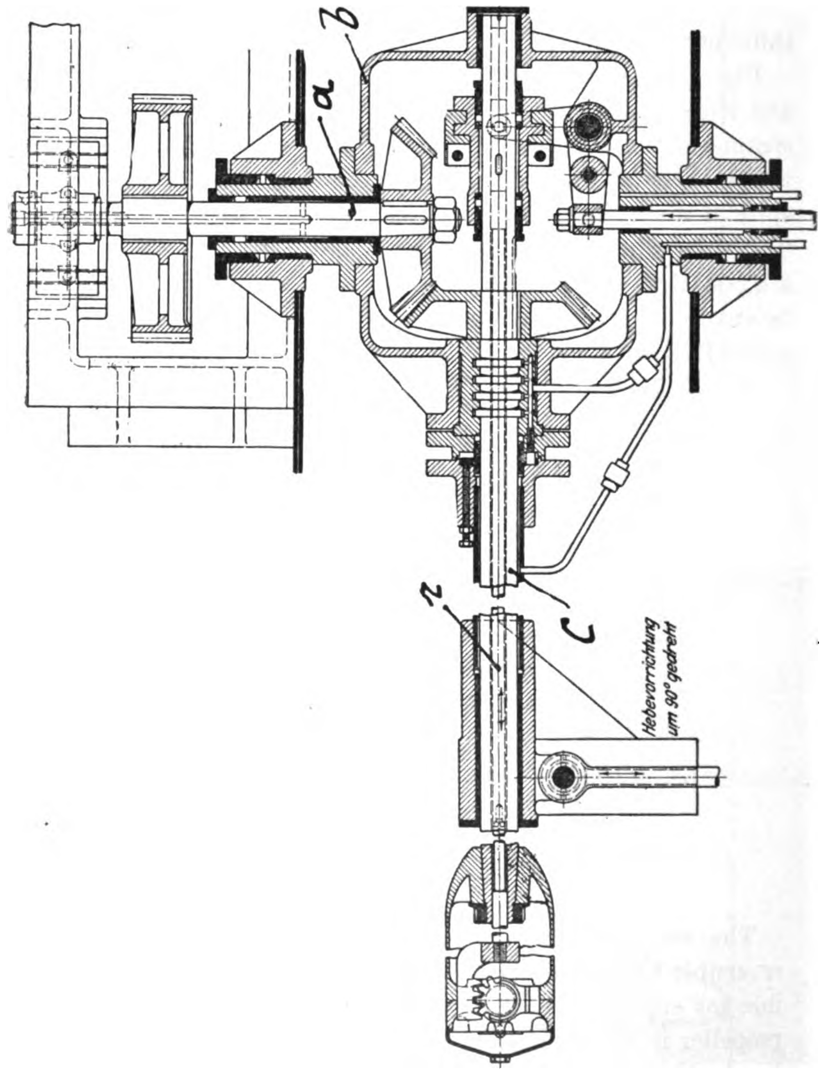


Fig. 7.

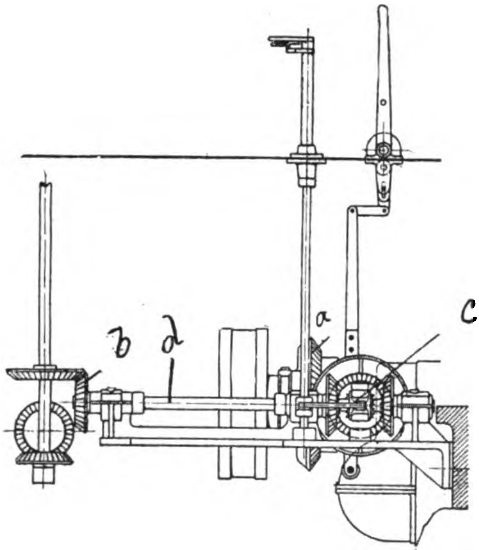


Fig. 8.

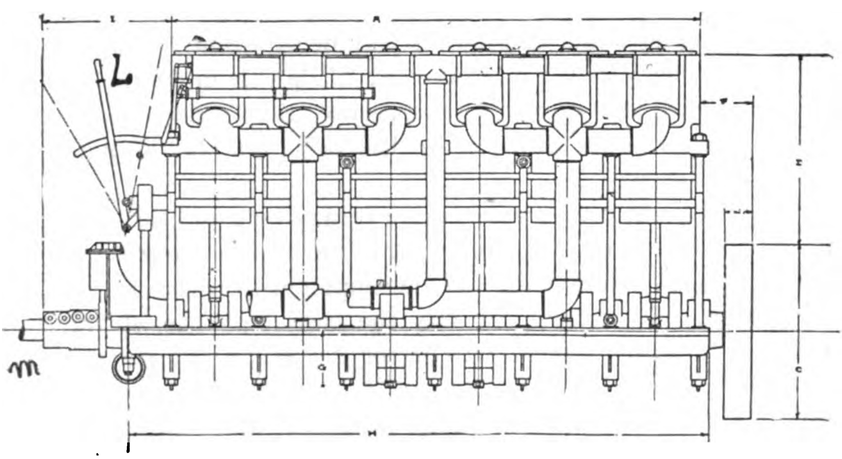


Fig. 9.

casing (b), by means of bevel wheels to the propeller shaft (c), as shown in Fig. 7. The thrust bearing acts on the two casing bearings, thus transmitting the engine power to the ship. The reversing of the wheel is done by a rod (r), inside of the propeller shaft, and the blades are turned by teeth, acting like

spur gears. The entire mechanism is enclosed and safe. The arrangement of lifting is shown in Fig. 10, at the stern, by means of a handwheel, screw and rod, at (m). The gas engine in this case is a standard, horizontal, stationary one, and belted to the main drive.

As soon as the power is higher than 100 H.P. it is often very difficult to move these reversing levers by hand alone. It was therefore required to install operating mechanisms for this purpose. Fig. 8 shows the vertical view of an arrangement whereby the power of the engine is used to reverse the propeller wheel. By a friction clutch (a), and gearings (b), and a claw coupling (c), those parts are connected, and act on the inside-shaft movable rod (d) for the blades.

Here I will note that in the first years of their development reversible propeller wheels were strongly objected to by marine engineers in general. But now, after about twenty years' experience on these lines, it may be said that most factories have very little trouble due to defect in weak construction or to wear. The greater portion of the complaints is due to ice or to running aground, and here it may be said that just such a propeller, with easily disconnected blades, is much more easily repaired, and at much less expense, than solid wheels. In the latter case the entire screw has to be renewed if only one blade is damaged.

In regard to the reversing question it may be noted that the number of direct reversible engines is increasing every year, and there is no doubt that great development in this important feature will follow, as was the case with the steam engine. The gas engine will never be considered perfect from the view of the Navy engineer, if it is not directly reversible. A good deal has been done already in this direction, however, and, besides some European concerns, there are some very good American makers coming in the market. The later makes are, in some cases, of the 6 and 8-cylinder type, and they have some advantages. Vibration is nearly eliminated, and, owing to the increased number of cranks, these machines run as smoothly as the finest balanced steam engines. The

flywheel and weight are greatly reduced, as the torque and strains are almost continuous. If desired, the engine can be operated without a flywheel. An elevation is shown in Fig. 9. Starting and reversing is as positive as in a steam engine, and the propeller is coupled at (m) direct to the engine shaft.

The operation is as follows: The valve motion is shifted by the lever (L) from a central point to the forward or reverse motion, and on opening the compressed-air valve, the engine, having no dead centers, instantly starts up. The compressed air is then shut off. A small air compressor on the engine keeps up the supply of air. The steel tanks hold enough air for starting 20 or 30 times without pumping. In the very large engines there are auxiliary air-pump engines, which are similar to the auxiliaries supplied with a steam outfit. A number of Russian torpedo boats, with 1,500 H.P., have been equipped with these famous engines.

Another method of reversing the motion of the engine is by advancing the spark, thereby causing a so-called "kick back." This feature is reliable enough for smaller boats. These engines with low compression can also be started ahead or astern at the will of the operator.

Like all other igniter operating devices, this one is equipped with devices for changing the time of ignition, according to the quality of the fuel, as well as according to the speed at which it is desired to run the engine.

In starting the engine the reverse lever is placed at a point which sets the igniters in such a way as to make the spark, after the piston starts on the downward stroke, in the direction it is desired. In reversing, the electric-battery main switch is opened, and the lever moved across the quadrant to the opposite side. The switch is closed after reversing.

As was natural, the gas engine was first introduced only in small sizes and in small craft. One of the earliest gas engines ever installed on water is probably the city-gas engine operating the floating bridge over the Rhine, in Cologne, since the year 1882. This was a four-cycle Otto engine of special design, with vertical shaft; this type of engine has since been

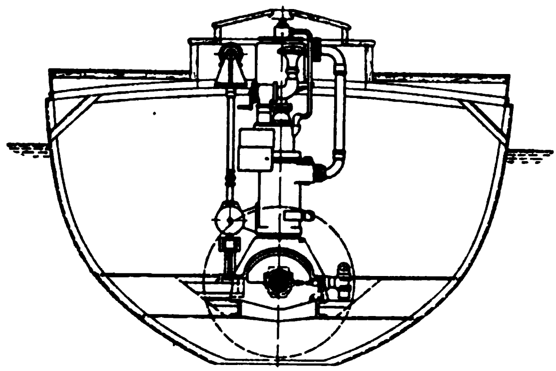


Fig. 10.

abandoned. The four-cycle principle, however, is leading to-day in marine gas-engine building. The two-cycle principle is not yet so well developed.

An early two-cylinder vertical kerosene engine of foreign make is shown in Fig. 10. It shows the lighter *Esse* with a 16-H.P. kerosene engine. The boat has a speed of 8 knots. The engine shaft is coupled to the propeller shaft, and the screw blades are reversible. The operating wheel is on deck to the right of the steering wheel. It is interesting to note the relative costs per ton, mile and hour. At a kerosene consumption of about .88 pounds per H.P. an hour, and a price

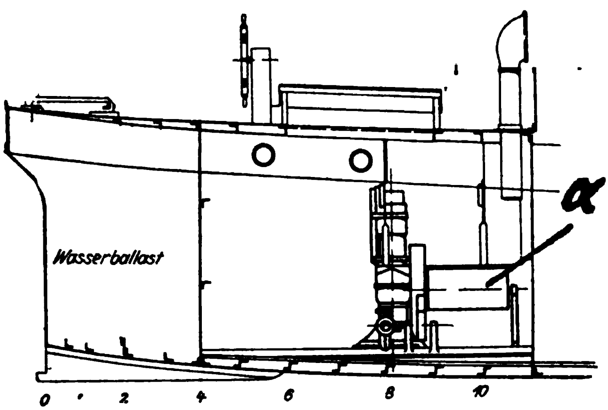


Fig. 11.

of 2.3 cents per pound, the cost per one ton, knot, hour, for 40 tons load was: $(16 \times 0.88 \times 2.3) : (8 \times 40) = .10$ cents. In reducing the speed to only 4½ knots and using only a 6-H.P. engine the cost would be: $(6 \times 0.88 \times 2.3) : (4.5 \times 40) = .067$ cents. At that time marine gas-engine practice was not so good as today, when it is known that there must be a proper relation of speed and load to obtain the best and most economical result.

In 1896 the *Petrolia* was equipped with kerosene engines. She was built as a sail boat, having kerosene tanks of 210 tons capacity. She had a length of about 100 feet, beam of 18 feet, and 7 feet draught. The engine developed 20 H.P. at 360 revolutions per minute; the arrangement was in principle the same as in *Esse*. The cost per knot, ton, hour, has been reduced to $(20 \times 0.88 \times 2.33) : (210 \times 4) = 0.048$ cents, with a speed of about 4 knots per hour. It must be noted, on the other hand, that the expenses for the wages of the crew have to be considered as well. A similar early towboat has 50 feet length, 10 feet beam, and 5 feet draught, and is equipped with a 20-H.P. kerosene engine. The speed is about 8½ knots per hour. The propeller wheel has two blades, and is 3 feet in diameter. There is a tank of about 40 cubic feet capacity for the supply of the engines, which is sufficient for about 100 hours' run, or about 850 knots action radius for only 40 cubic feet of bunker capacity. The entire machinery equipment requires about ⅐ of the ship's length. An equivalent powerful steamer equipment would need about ⅓ of the ship's length. With a coal consumption of 4½ pounds per H.P. per hour, which is certainly fair, the comparison shows as follows: the above 40 cubic feet would contain about 2,400 pounds of coal, which is sufficient for only 26 hours' run, or about 221 knots action radius. Considering, however, the costs per H.P. hour, the comparison would stand as 1 to 2 or 1 to 3 in favor of the steam power. But as the action radius stands as 4 to 1 or 5 to 1, so this example shows characteristically the great value of small motor boats as auxiliary craft to be kept on board of large men-of-war, and to be used in cases where quick action is required.

The arrangement of a foreign 4-cycle gasoline engine driving twin propellers is shown in Fig. 11. The two propeller shafts are each driven by a pair of open and crossed belts from a main pulley connected to the engine shaft. This arrangement, however, has proved to be not very satisfactory, as the belting on such short distances of the shafts did not work well.

I will state here that it is advisable, for various qualities of liquid fuels, to use various arrangements of devices and methods to form the mixture in explosion chambers. Obvious, however, as this rule might be, there are numerous mistakes made. It has not been, and never will be, practicable to use the same devices for various fuels. One physical rule is to be observed for all similar combustions: separate the fuel in as fine and small particles as possible, and mix it as thoroughly as possible with the air. One of the best authorities, C. Stein, Germany, in this respect, says: "It is nonsense to attempt to build all marine combustion engines of a standard type. Each experienced engineer must know that the conditions in service for pleasure launches, tow-boats or naval purposes are entirely different. It would be certainly agreeable for the manufacturer to use one pattern for all systems, but it is wrong, as it will never suit the customer."

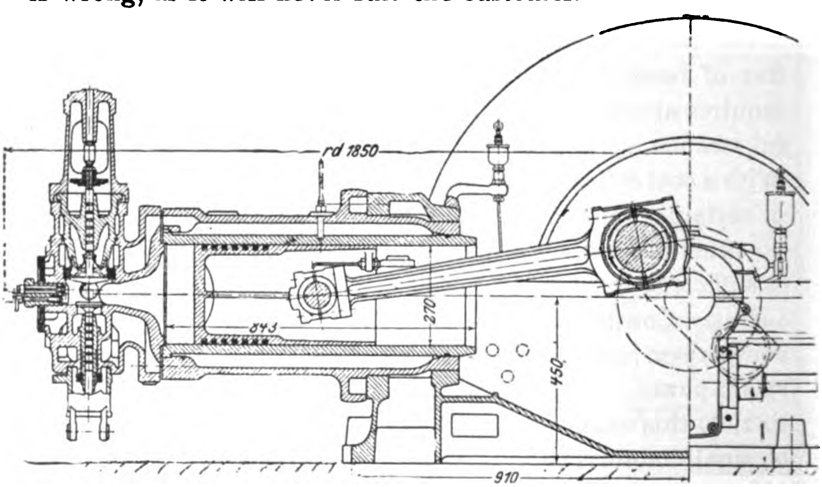
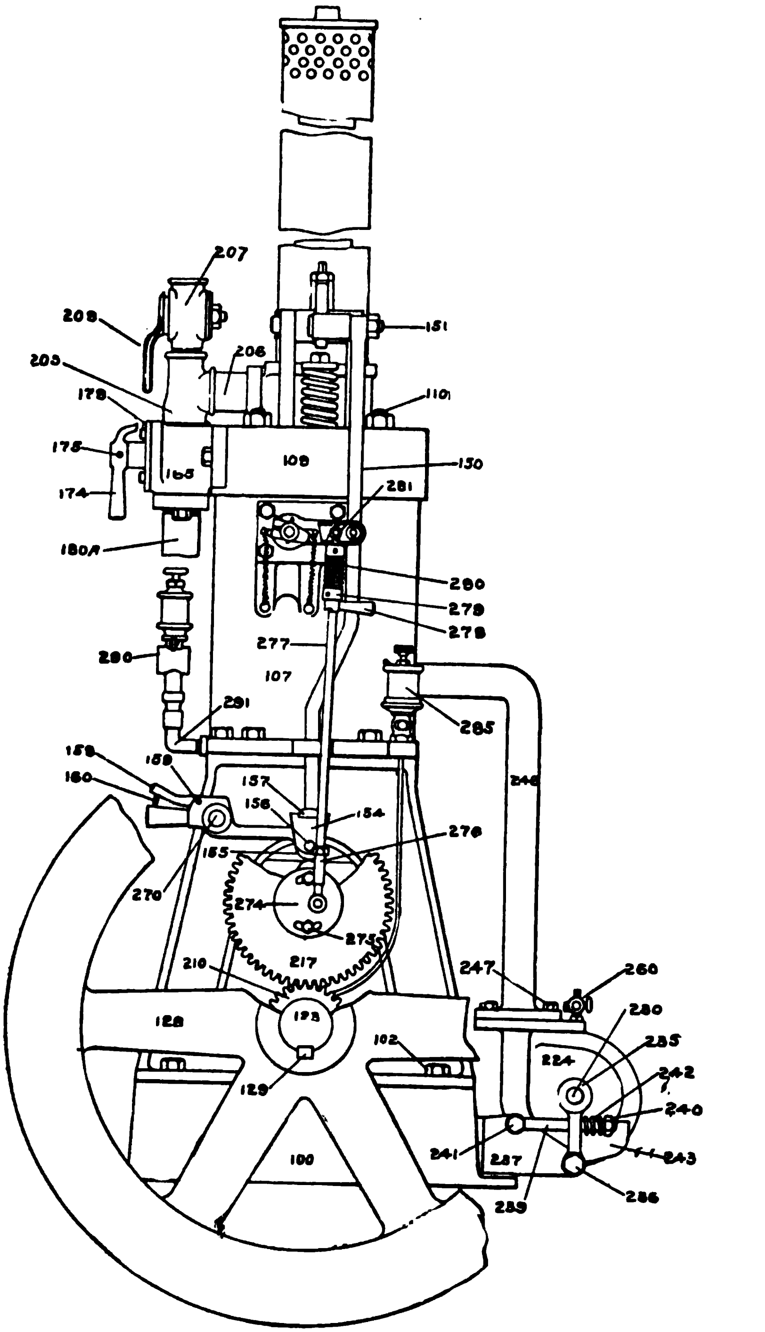


Fig. 12.



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100

Fig. 13.

Fig. 12 shows a horizontal marine engine with opposite cylinders. This pattern is specially suitable for high-compression engines, and is to be made lighter or heavier, according to the fuel. Valves and igniters are mechanically operated, and the piston can be drawn, towards the crank shaft, without disturbing any other part.

Fig. 13 shows an American engine of very well known make. It is based on the 4-cycle principle and has the following important features: A check valve in the air-suction, automatically-working suction valve, and mechanically-operated exhaust valve; an independent throttle arrangement for each cylinder, operated by hand as well as by a governor, a spark-timing device, and a hand-speed controller for the governor. It is a specially good gasoline engine, and the air is heated by the exhaust before starting the mixture process. The exhaust can be made upright, as shown in Fig. 13, or submerged, but is provided in the latter case with an air check. The cooling water is furnished by a rotary pump driven from the main shaft, and an air compressor for the required compressed air for whistling and starting, is attached. The electric spark is obtained by battery or dynamo. When distillate is to be used, there must be two fuel tanks—one small gasoline tank and a larger one for the distillate. The engine must be started with gasoline and should run long enough to warm the air heater sufficiently to vaporize the distillate. There is a reversing coupling for the propeller, and the operating lever is similar to a steam-engine arrangement. Larger engines are built on the same principle, but they have a mechanically-operated suction valve. The main objection to this engine is the inaccessibility of the cylinder heads for repairs; an advantage, however, may be said to exist in having both valves vertical in the cylinder head.

A very interesting type of a 4-cycle engine, of foreign make, is shown in Fig. 14. It is now built up to 600 H.P. in one unit of 4-cylinders, with only 120 revolutions. The single-acting piston was accepted, and it reduces the weight of the engine.

High compression was preferred, as it is built as a special marine producer-gas engine. I will say here that I perfer this system for very clean liquid fuels, but not for producer gas.

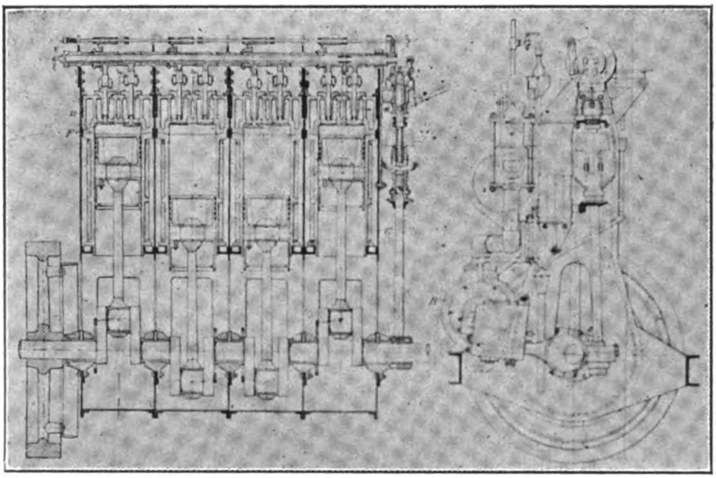


Fig. 14.

The engine shown has no cages for the valves and it is, therefore, necessary to remove the entire cylinder head if the valves must be cleaned. The cylinder hangs down from the heads, thus allowing a free expansion, and the strain of the explosions is taken up by a steel-plate structure. These are very remarkable features, but only time will prove if they are really good for marine purposes. The countershaft runs across the heads and operates the valves by single links, and the igniters are timed by means of a rod lying in this hollow countershaft. This arrangement, I think, is especially preferable in smaller engines, as it allows an easy accessibility, and observation of the operating parts. For larger engines, however, it will probably be objectionable on account of wear and noise.

For operating a boat in shoal water, however, it is necessary to provide other means for propulsion, and stern wheels are therefore preferred. Side wheels are used principally for ferry

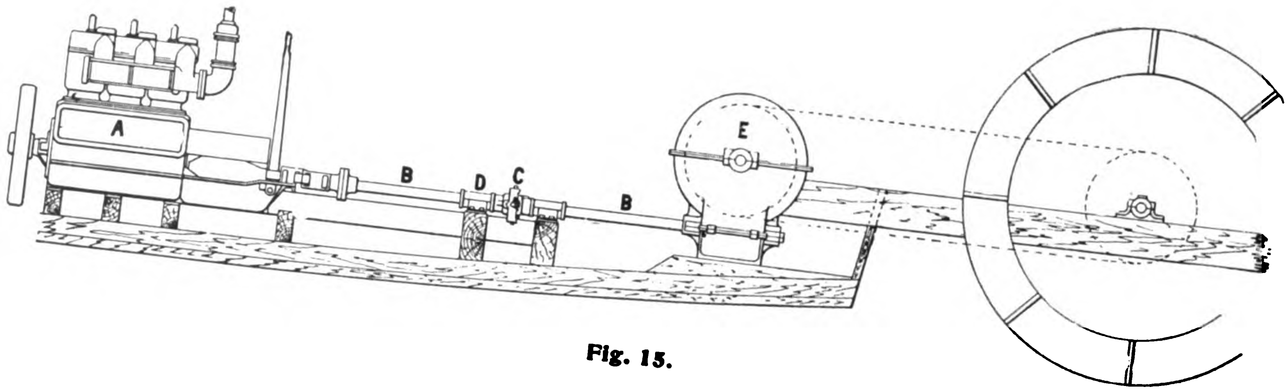


Fig. 15.

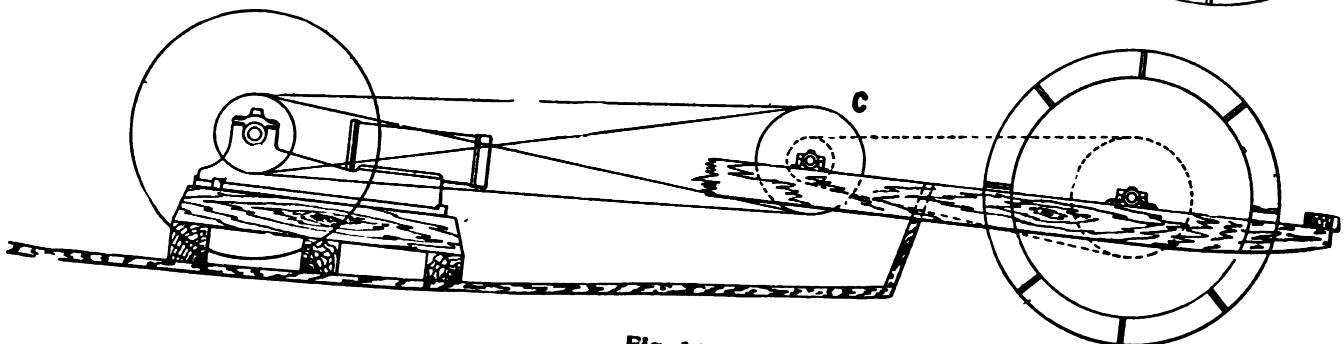


Fig. 16.

boats where it is necessary to provide a clear space through the center of the vessel, and to make a landing with either end at the dock or slip. Fig. 15 shows the arrangement of a vertical gas engine, where the power is transmitted to the stern wheel by means of a worm-gear and sprocket-chain connection. Fig. 16 shows the installation of a standard horizontal stationary engine in a boat, transmitting the power to the stern wheel by means of straight and crossed belts, and tight and loose pulleys, to a countershaft F; thence to the wheel H by a sprocket-chain drive L. The reversing is done in case 41 by a reversing clutch, and in 42 by the crossed belts.

A characteristic American construction is shown in Fig. 17.

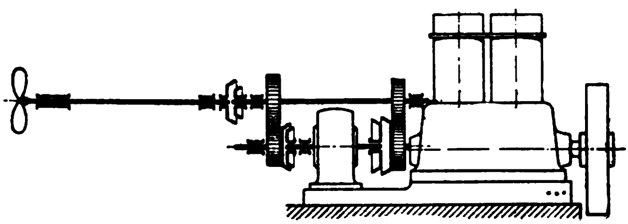


Fig. 17.

It is a 200-H.P. kerosene engine, vertical, with two cylinders, and especially designed for submarine-boat services of the Holland type. Directly connected to the engine shaft is an electric generator, thus taking advantage of the high speed. The screw propeller shaft is driven by spur gears. On the surface the engine runs the propeller or charges the storage batteries. Below the surface this generator is switched in as a motor and drives the propeller wheel from the battery. This arrangement is also very convenient for starting the kerosene engine by the electro-motor.

Another citation of authority, Rankin Kennedy (England), on marine engines is as follows:

"The moving parts of the engine should be the same as those of a standard marine *steam* engine, and there is no reason why they should be otherwise. To box them up, *out of sight*, in a crank case, would be *absurd*, although among oil and kerosene motor-car engine makers this proceeding is *fashion-*

able. When we come to marine (navy) engines, the conditions are so far different that *open* engines must be insisted upon, and other departures from the conventional practices must be made when marine propulsion by internal combustion engines is *seriously* considered, either for commercial or naval purposes. The present craze is to install too light weight motor-car engines. This is not good marine engineering. As a matter of safety the marine engine is far too important to be trifled with, and experience with the steam engine is not to be lightly thrown aside for the whims of the fashionable motor-car builder. Hence in designing internal-combustion engines for business purposes the closed motor-car model is useless, and engines modelled more in the shape of steam engine practice are necessarily to be adopted."

That *reliable* gas-engine construction was going the same way as stationary engines may be proved by a few words and drawings as follows:

Fig. 18 shows a "Korting" two-cycle engine, driving a horizontal blower engine for blast furnaces. Is there any service in mechanical engineering where *reliability* and *durability* is of more importance than in the blast furnace practice? These engines must run years and years, without giving any considerable trouble, and yet we see the steam engine replaced by the gas engine.

Fig. 20 shows the "Deutz" four-cycle engine, and if the piston rods and valves did not show "water cooling" it might be easily mistaken for a modern steam engine.

Fig. 19 shows the "Cockerill" tandem gas engine, especially as twin-tandem, giving any degree of steadiness, and satisfying the most severe conditions of running alternating generators in parallel.

Fig. 21 shows another big engine, "Oechelhauser," two pistons, single cylinder, two-cycle, well known on account of its claim as a valveless engine, specially preferred for heavy services, like rolling mill and blower engine driving.

Fig. 22 shows the "Nurnberg type" driving a blower engine, but with specially strong claims as being an engine of good regulation, and steadiness, and famous workmanship.

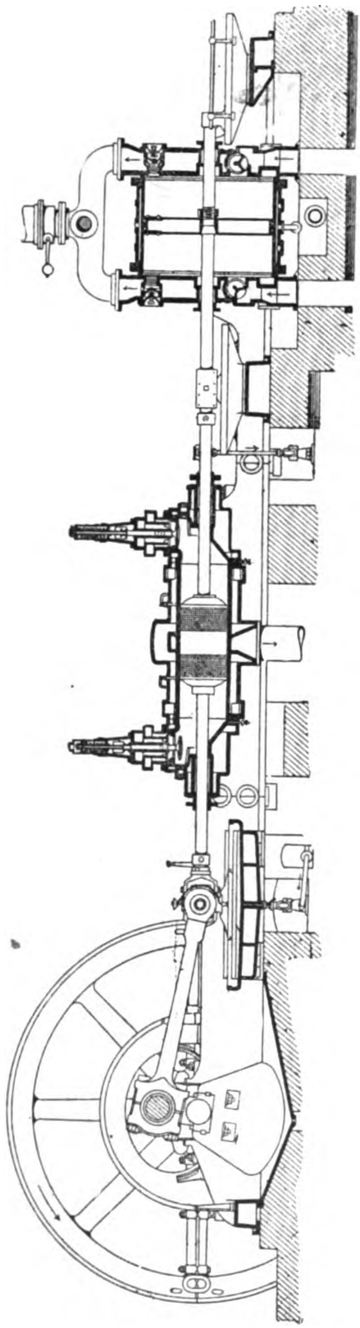


Fig. 18.

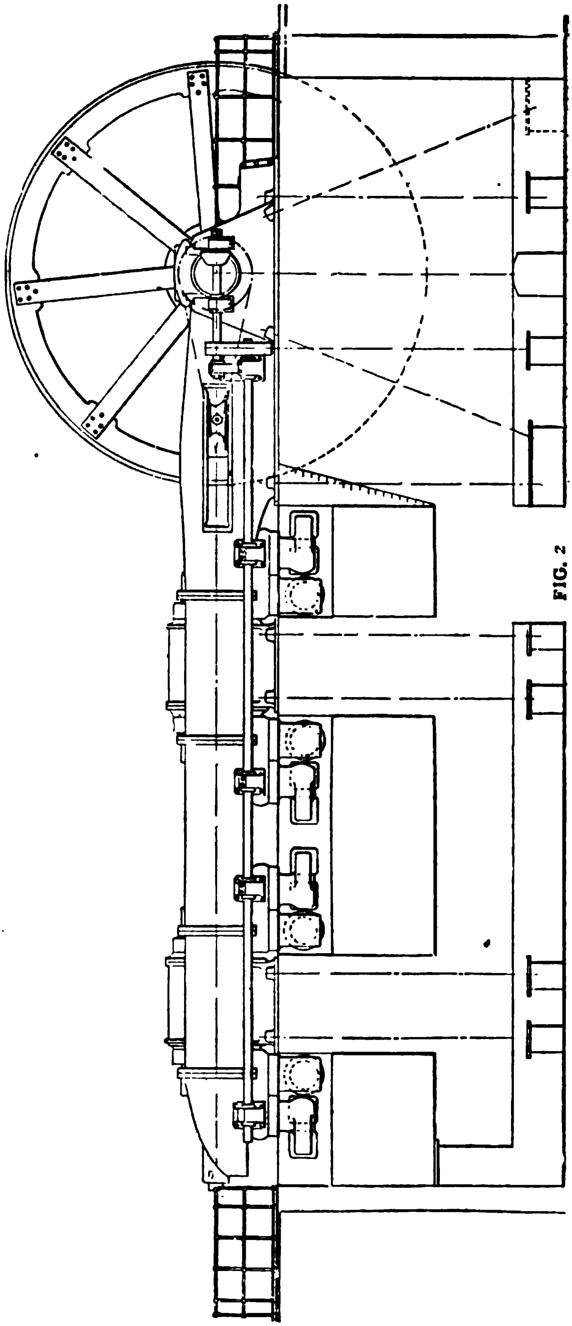


FIG. 2

Fig. 19.

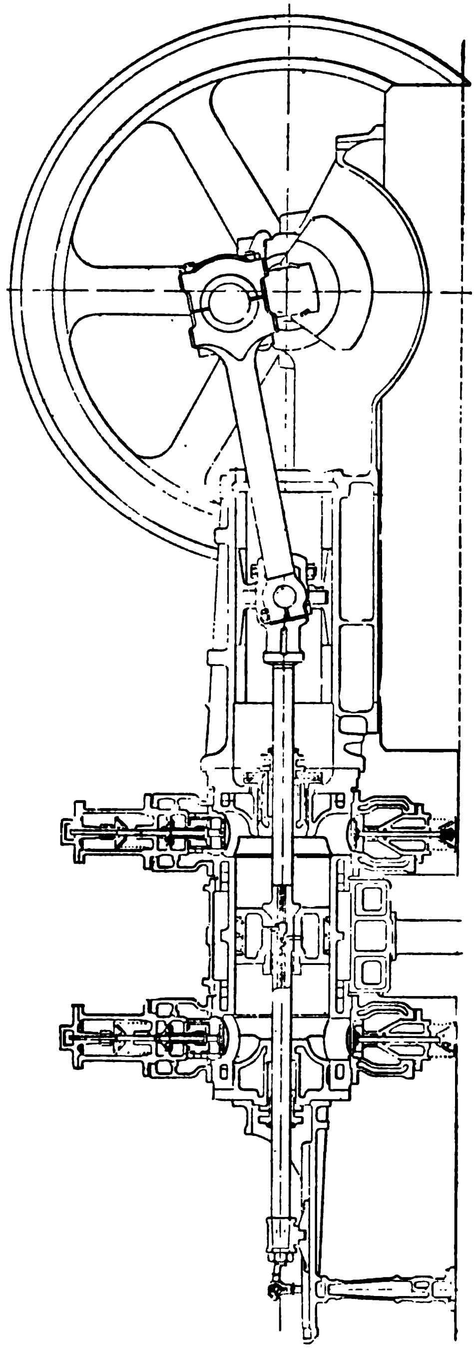


Fig. 20.

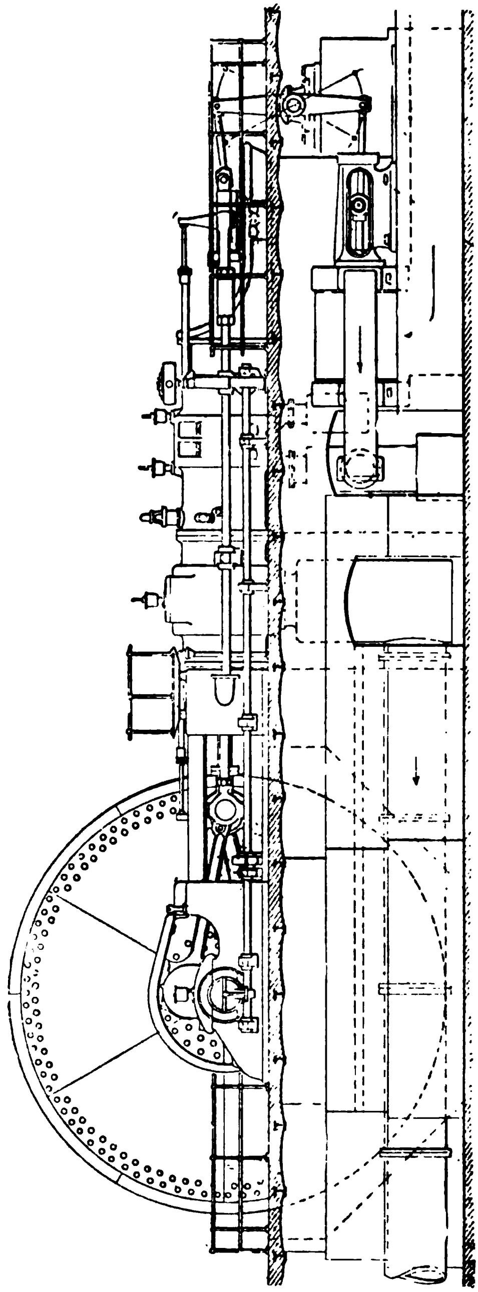


Fig. 21.

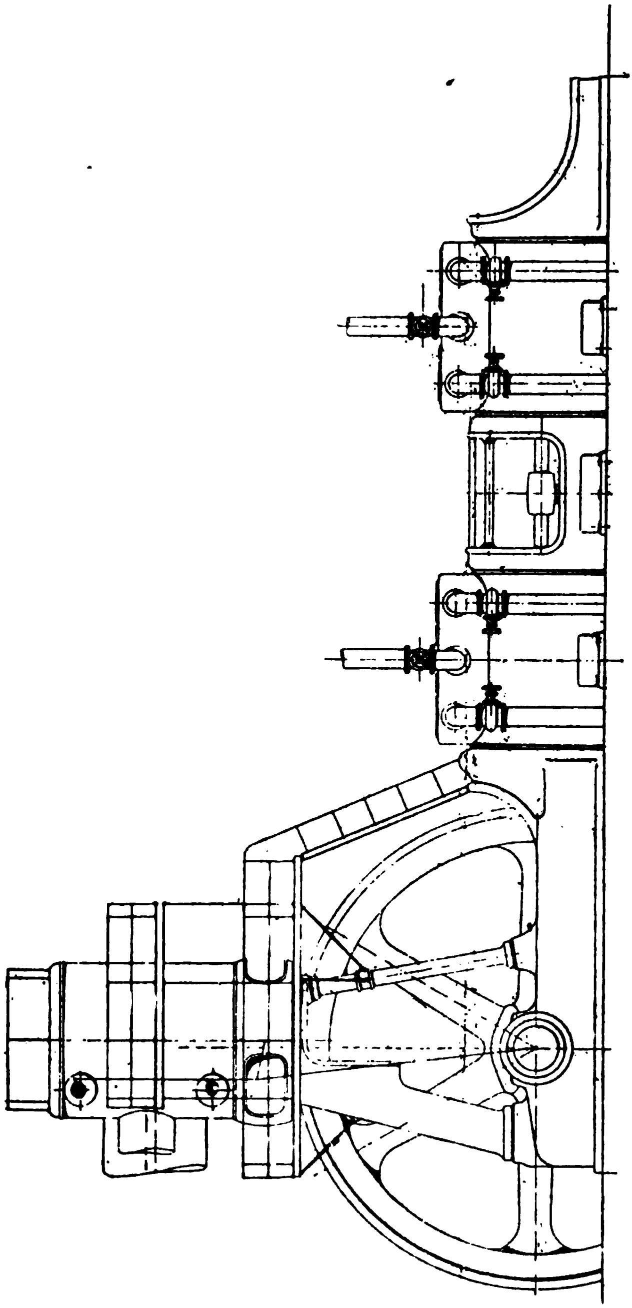


Fig. 22.

This greatest development in the history of mechanical power was giving confidence to the marine engineer, and the latest step in this direction was the installation of "suction-gas plants" in marine vessels.

Before entering, however, this field, there may be said a few words more on modern power boats running on liquid fuels. It might be noted that in the last few years there has been a tendency to construct the hull of the vessel to suit the conditions of the internal-combustion engines, and this type of boat is very different from those driven by steam. Therefore it would be advisable to follow the practice of steam engineering so far as to furnish not only the gas engine, but also the boat and other equipments. Usually, a shipyard furnishes both the hull and the entire machinery. Why not do the same in gas engines? This is done now, more and more, and a new type of hull construction has resulted. Tests with small models were made by Mr. Kretschmer, of Berlin, and gave the following interesting results: The model was made according to the dimensions of an up-to-date ocean liner, about 630 feet long, 75 feet beam and 24 feet draught, and having a tonnage of 16,800 tons. The corresponding results of this model with the new tetrahedral form are given in this table:

Speed and knots per hour:	Required horsepowers:
16.3	4,150
19.6	7,450
22.8	13,120
25.0	19,000
26.1	22,500
29.7	37,900

A boat of French make of this kind is to be seen in Fig. 23, *Titan II*, and Fig. 24, the *Rapee III*.

The widest framing is at the stern. The bottom portion of the hull is ball shaped and the bow is entirely flat. In motion, the bow rises out of the water, which results in a decreased water friction. Bow waves are not noticeable, and there is only a slight swelling of the water on both sides of the stern. The length over all of both the above mentioned boats is about

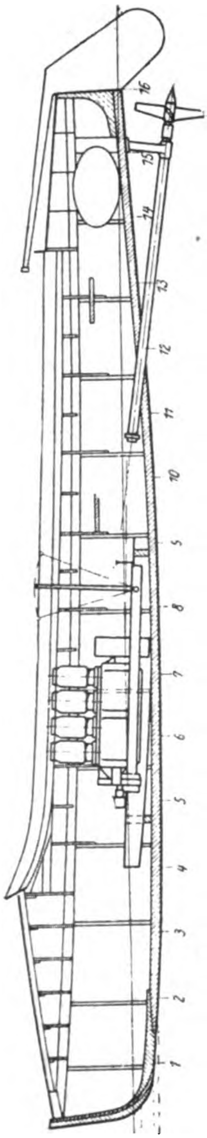


Fig. 23.

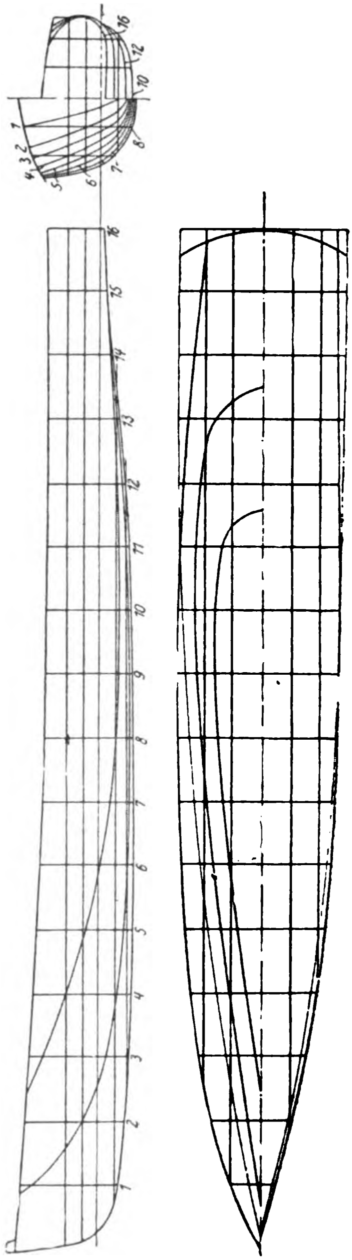


Fig. 24.

26 feet and 3 inches, and the length on water line is 25 feet 10 inches; the greatest beam is 5 feet; the draught, at rib, is $6\frac{1}{4}$ inches, at rib 8 is 11 inches, and at rib 16, at stern, is 22.5 inches. The tonnage is 1,158 and the largest cross-scantling section is 2.15 square feet. It was necessary to drive the propeller shaft by a universal coupling, in order to get the blades as deep in the water as desired. The reversing is done by a reversing friction coupling. The oil engine for *Titan II*, has four vertical cylinders of $5\frac{5}{16}$ inches in diameter and $5\frac{1}{8}$ inches stroke, and the number of revolutions is 1,415 per minute, giving about 54 indicated horsepower. The speed was 19.2 knots per hour. The propeller had three blades with $18\frac{1}{8}$ inches pitch and $20\frac{3}{4}$ inches diameter, giving 12.3 per cent. slip. The *Rapee III*, has an 80-horsepower, 4-cylinder oil engine of $6\frac{3}{4}$-inch stroke and 960 revolutions per minute. The obtained speed was 21.8 knots with a screw of three blades, 26 inches diameter, $30\frac{3}{4}$ inches pitch, and about the same slip as above.

I will state here that the weight of an American auto-launch gasoline engine of 25 horsepower, with flywheel, is only 300 to 450 pounds, of the design as shown in Fig. 17, having four cylinders. Even these weights are found to be reduced to ten pounds per horsepower, as I saw on some French makes in St. Louis World's Fair.

A speed of 26 knots was obtained recently with an English craft of only 40 feet length, 5 feet beam and 8 inches draught. The entire weight was $3\frac{1}{2}$ tons, of which about 1.75 tons was for the complete mechanical outfit. There are two gasoline engines, as shown in Fig. 25, each having four cylinders and developing 60 horsepower, making 120 for both. The dimensions are: cylinder, $6\frac{1}{2}$ inches diameter; stroke, 6 inches. The center line of propeller and engine shaft, have a considerable slant, in order to get the blade as deep as possible in the water.

A very small but interesting torpedo boat is shown in Fig. 26. It is of English make and has a speed of 18 knots per hour. The length over all is 40 feet, the beam, 6 feet 4 inches, and the draught $30\frac{3}{4}$ inches. The capacity is $4\frac{1}{2}$ tons. The engine has been placed forward, as the stern has to be free for the

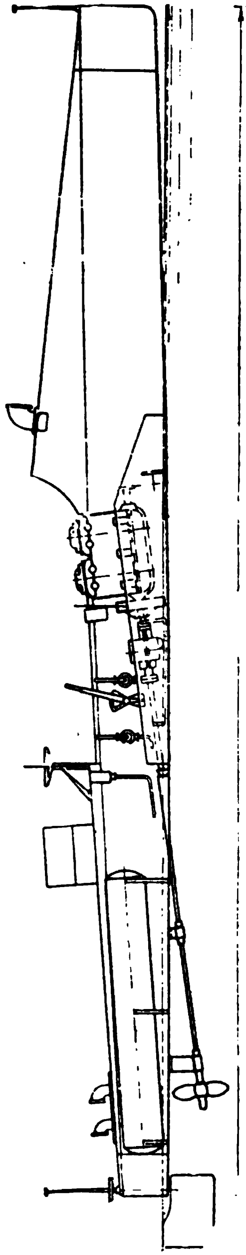


Fig. 25.

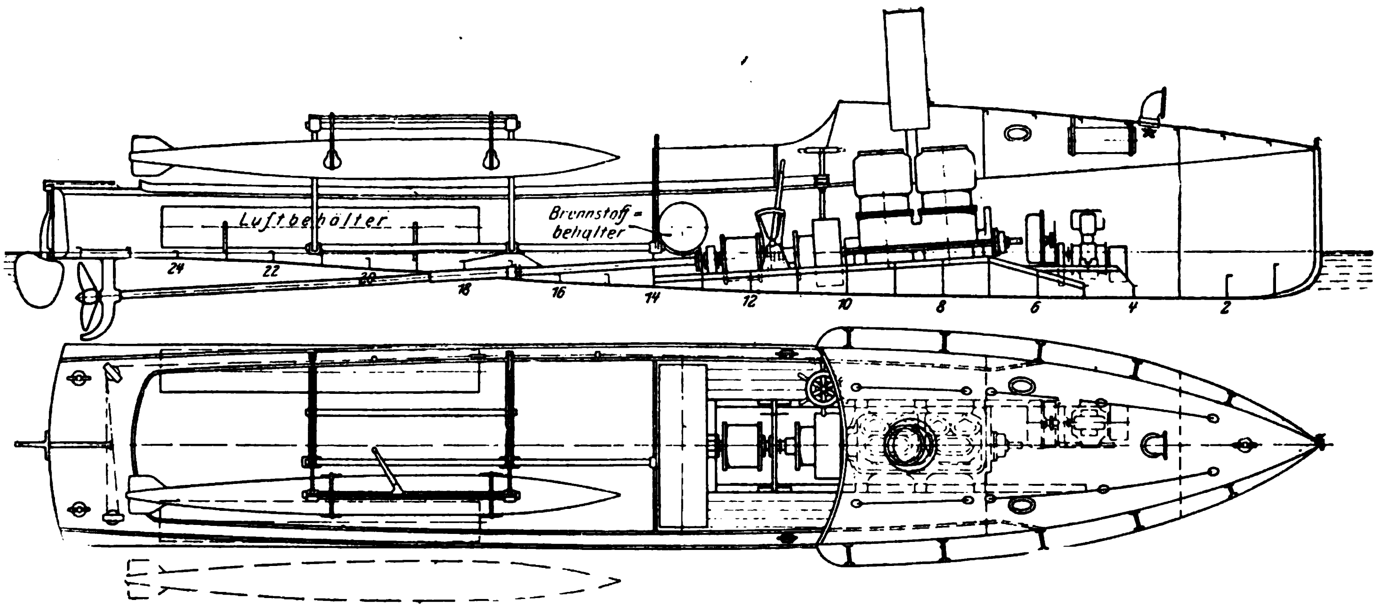

Luftbehälter
Brennstoff-
behälter
24
22
20
18
16
14
12
10
8
6
4
2

Fig. 26.

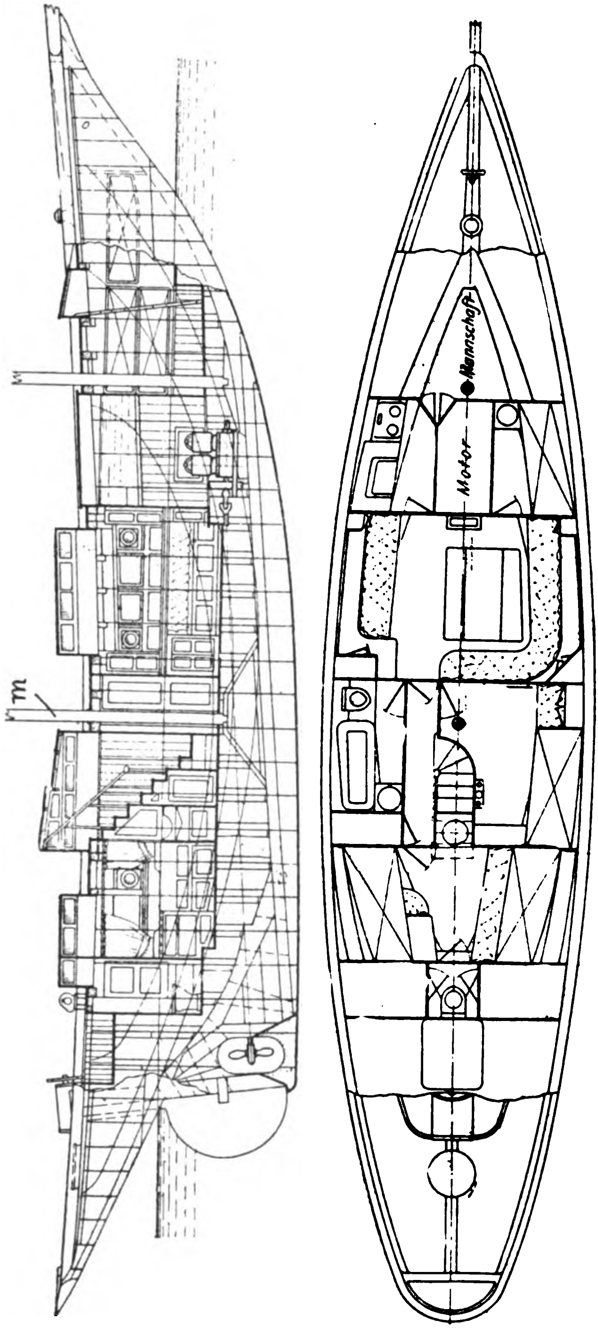


Fig. 27.

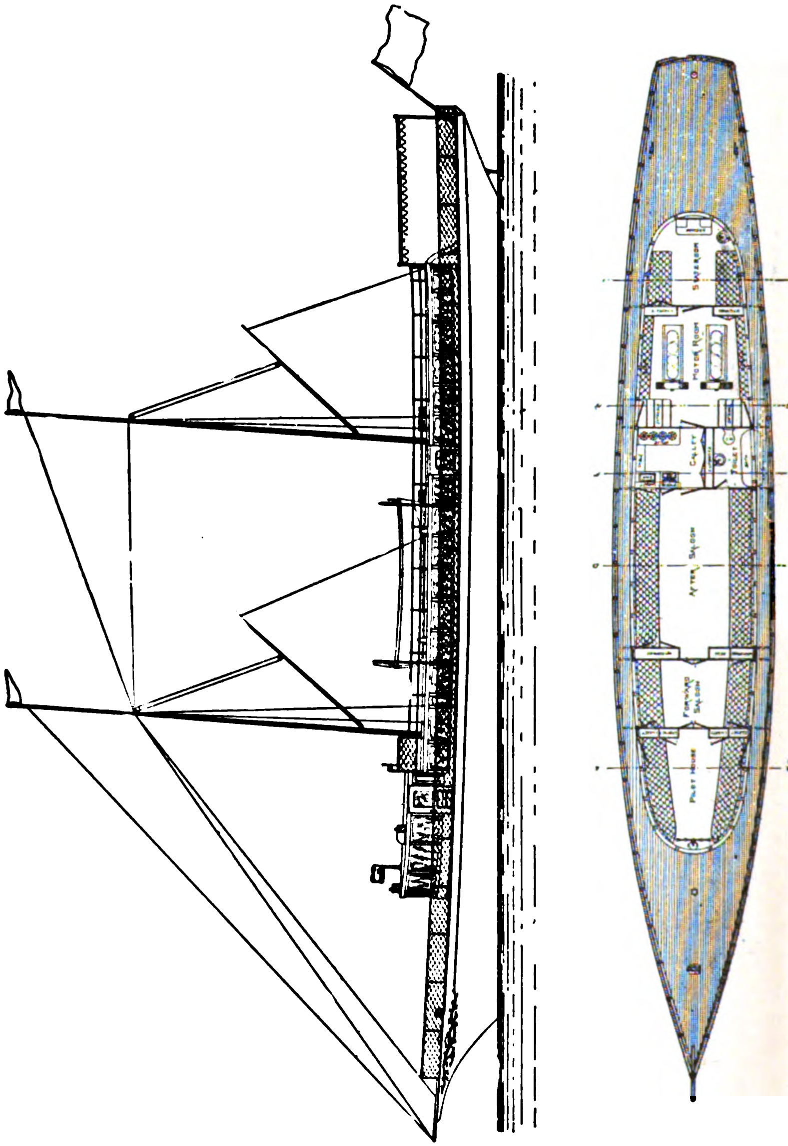

Stateroom
Motor Room
Galley
Toilet
After Saloon
Forward Saloon
Pilot House

Fig. 28.

torpedo outfit. This engine has four cylinders of 8 inches diameter and 8 inches stroke, and makes 900 revolutions. The rated indicated horsepower is 120. The entire machinery has a weight of 2,750 pounds.

The air for starting is compressed in a separate, vertical, 6-horsepower gasoline engine. Reversing is done by a reversible coupling.

A German pleasure launch equipped with a complete sailing outfit as well as an auxiliary gasoline engine is shown in Fig. 27. The engine has 30 horsepower and runs with 400 revolutions. For fire protection, all walls of the engine room are covered with lead, and a fan exhausts all foul air and gases directly into the atmosphere. The propeller wheel is connected with the engine by means of an universal coupling and the shaft crosses the foot of the main mast M. The speed is 7 knots. The dimensions are: 65 feet length over all and 45 feet on water line; 14 feet beam, and 7 feet draught. The entire area of sails is about 800 square feet.

A 40-foot cabined boat of American make has about 10 feet beam and full accommodations for an extended cruise. The pilot house and saloon are well arranged in front of the after engine room. The boat is generally equipped with a three-cylinder 30-horsepower gasoline engine of about 1,200 pounds weight, and may cost, including the reversing coupling, from $1,000 to $1,200. A 90-foot cruising yacht of fine design and practical equipment, shown in Fig. 28, is of well-known American make. It has about 12 feet beam and 5 feet draught. There are twin screws, each with a reversible coupling connected to a four-cylinder gasoline engine. There is also a complete sailing outfit.

Before closing this chapter it might be interesting to notice one of the recent improvements in propeller-wheel design. For internal-explosion engines with four-cycle combustion, a wheel is particularly desired which reduces the vibrations of shaft and engine. Graf Rudolf Westphalen designed the propeller shown in Fig. 29. This is a wheel for a small cruiser of the German navy. This vessel is 320 feet long, 40 feet

beam, and 15 feet draught, with a displacement of about 2,700 tons. The diameter of the screw is 9 feet 10 inches. The engines develop 8,000 horsepower, and the speed at 160 revolutions is about 21 knots per hour. Figs. 30 and 31 show clearly the difference between the vibrations of an ordinary

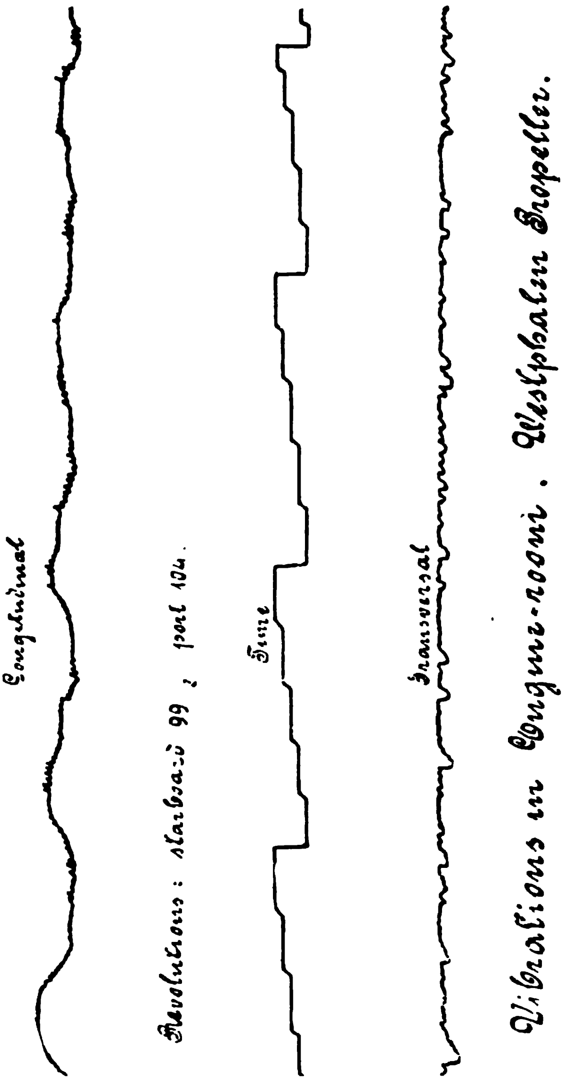


Fig. 30.

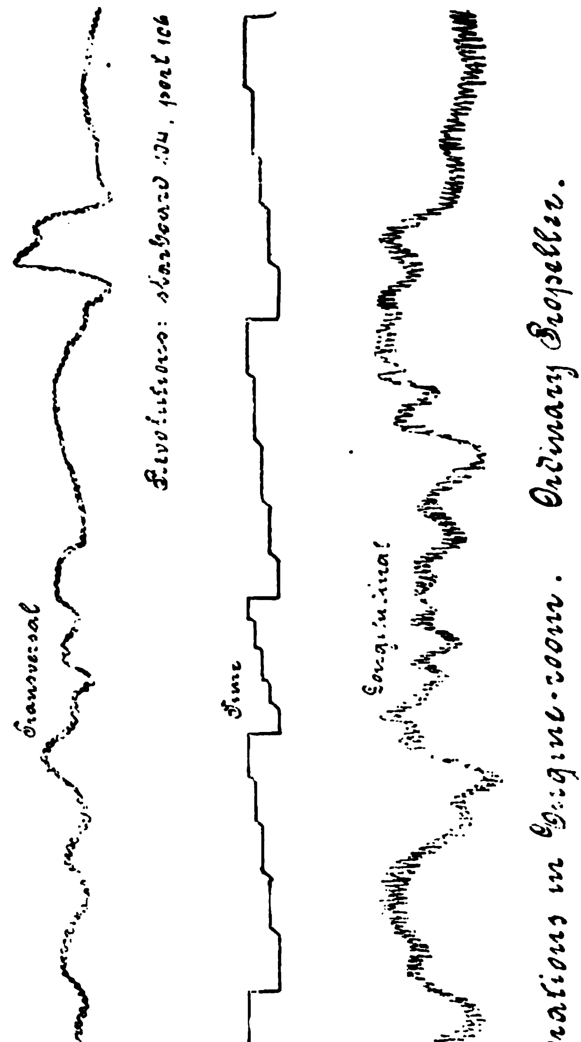


Fig. 31.

screw and the Westphalen propeller, as taken in the engine room with an indicator, the so-called "pallograph" of "Schlick" design. Fig. 31 gives the vibrations in the engine room of the North German Lloyd steamer *Seeadler*, equipped with a propeller of general type, having three blades directly

attached to the hub. Fig. 30 is equipped with Westphalen's propeller, also having three blades. It is of special interest to compare the diagrams of the longitudinal and transversal vibrations.

Now, near the end of my paper, it might be of interest to show some cuts and give some descriptions of coal gas producers of later improvements and designs, as they are in general use for driving stationary gas engines.

The efficiency of a producer plant is about 80 to 84 per cent. Besides the good adaptability of suction-gas plants to 4-cycle engines, and pressure plants for 2-cycle engines, it may be noted that the latter are more suitable for low grades of fuel. The suction-gas producers of today are considered especially adapted to anthracite, coke or charcoal. The efforts of many inventors have, therefore, been to construct suction-gas plants suitable for bituminous coal; I have succeeded in operating one of my own design running on all kinds of soft coal or lignites, even using wood, shavings, corncobs, etc.

All these producers are equipped with vaporizers or economizers for saving the heat of the escaping gas in order to reduce the fuel consumption. This gas is very hot, and one pound of fuel yields about *four* pounds of gas at a temperature of about 1,470 degrees Fahrenheit. The specific heat being taken at 0.23, and allowing 370 degrees Fahrenheit for the gas to enter the scrubber, we have about $1{,}100 \times 4 \times 0.23 = \mathit{1{,}012}$ B.T.U. for evaporating steam. If the boiler has, in this case, about 80 per cent. efficiency, and one pound of steam requires 1,146.6 B.T.U. latent heat, then this heat is sufficient to make $1{,}012 \times 0.80 : 1{,}146.6 = 0.70$ pound of steam. The steam costs, therefore, do not exist in suction-gas plants, showing their superiority over pressure producers.

Fig. 32 shows the general arrangement of a Deutz suction-gas plant. A is the ashpit, B the producer, D the upper coal magazine, L the chimney valve, and I the gas cleaner or scrubber. K is a gas dryer or purifier.

The wet-scrubber is filled with coke, and water is sprinkled in from the top. The dry-purifier, generally, is filled with

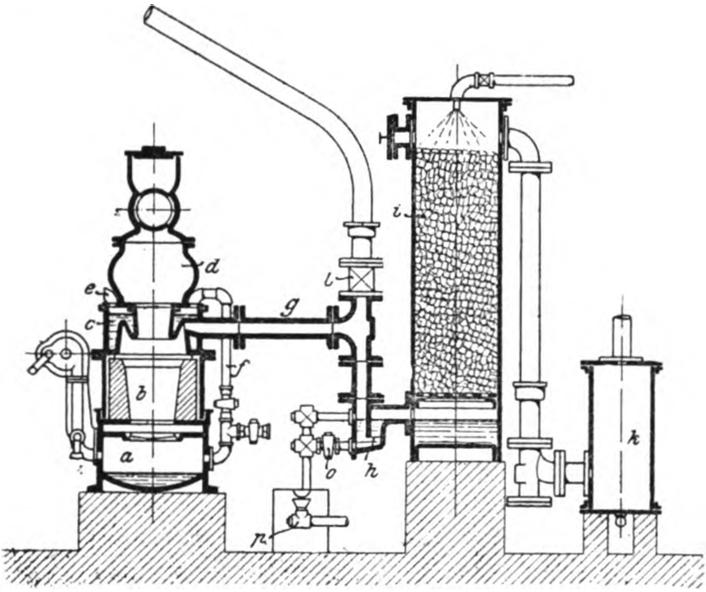


Fig. 32.

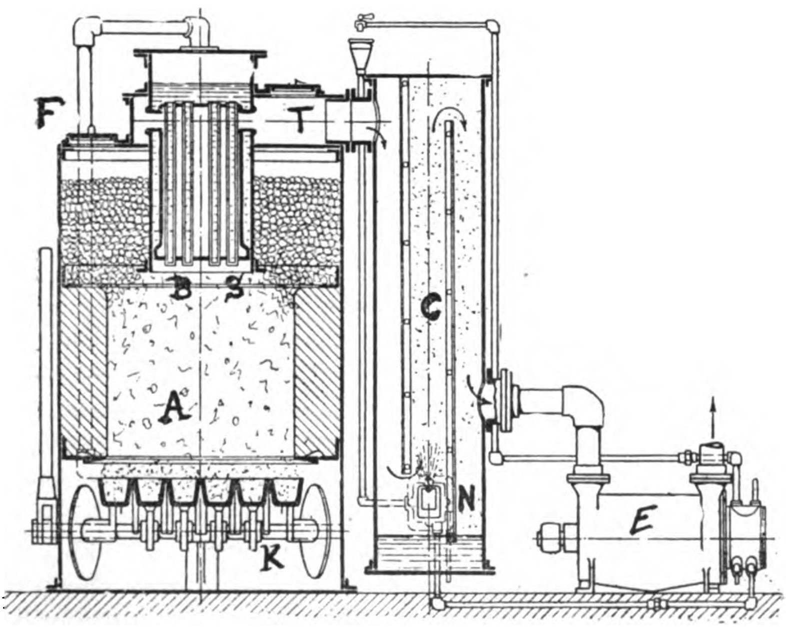


Fig. 33.

sawdust or excelsior in order to take out the fine water particles carried over from the scrubber with the gas. A vaporizer is shown on top of the producer at C, and the steam thus made by the heat of the gas is passed in the lower end of the producer by pipe F. The amount of water entering this vaporizer is, in up-to-date plants, governed according to the load of the engine. In A are flat grates, and I prefer the latter in such cases where a clinkering coal is used.

A special producer for marine purposes, system "Capitaine," is shown in Fig. 33. The producer A is of general construction, but the gas is drawn from the center, very near to the hottest zone, and escapes through T.

The scrubber C is equipped with fine nozzles N, for water sprinkling in the lower end. The coal is fed around the gas pipes S, and center boiler B through the holes F, generally closed with plain covers. There are no stationary grates in this producer, but rocking ones at K, in the ashpit.

The gas, after passing the scrubber, is mechanically cleaned by a tar extractor E of special design, and very high rotating speed.

When it is not desirable to use salt water in producers directly, a special salt-water vaporizer also of "Capitaine" design, Fig. 34, is recommended. In order to avoid the depositing of the salt in the vessel V, there is a continuous current of sea water going through. The percentage of salt, therefore, never goes higher than desired. The fresh water enters at A, passes around through the spiral pipes T, and enters the vaporizer V, in the lower end at B. The overflow is at C, in the upper part, and S is a circular boiler around V, from which the *thickened* water drains at Z. Thus the lost heat is used for a preheating of the fresh water. The water level is kept at a height W, and the steam escapes at P. The hot gases enter through M, and discharge through N, after passing downward through a number of tubes N. I will mention here that it is possible in this way to provide distilled water for drinking.

In closing this paper I will show a foreign boat, operated by an anthracite suction-gas plant. The great superiority

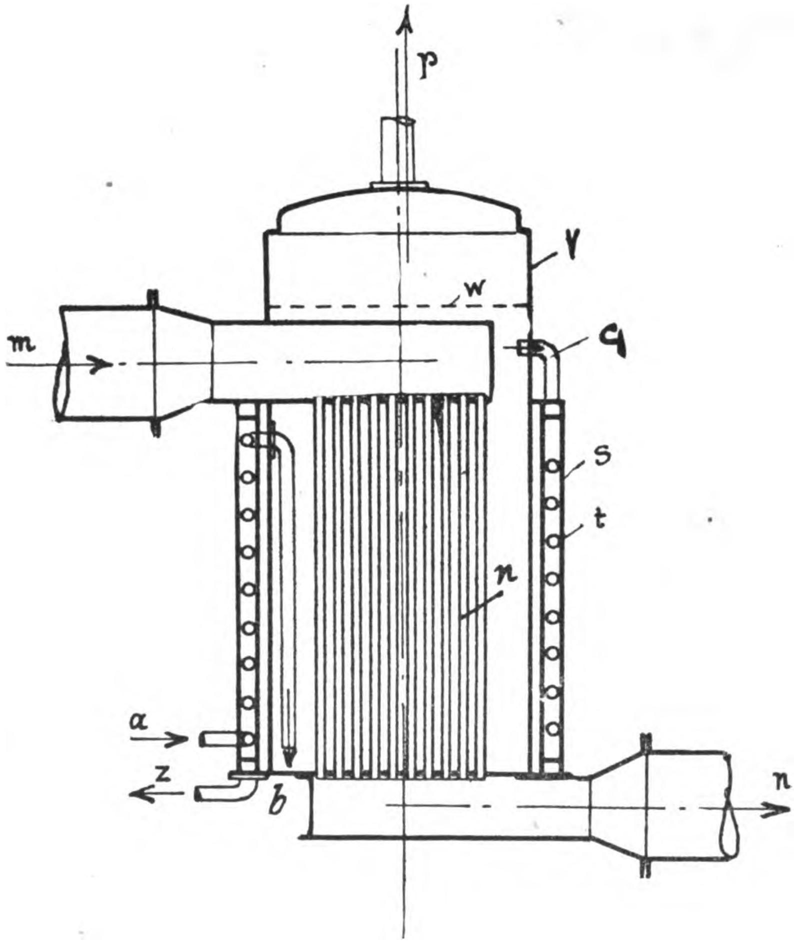


Fig. 34.

over steam outfits is to be found in saving of fuel and men for attendance, and, in comparison with oil engines, in avoiding fire danger by explosions, as well as cheaper fuel costs and saving of bulk.

The first suction-gas boat, *Haldy*, has been running for about 5 years, and the owners are very well satisfied with her. Deutz was soon building more of those boats, and his second

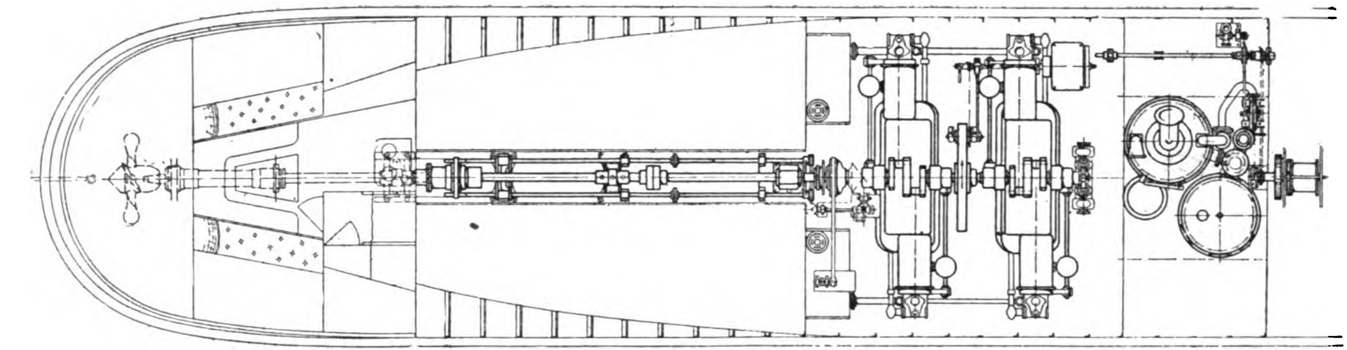


Fig. 35.

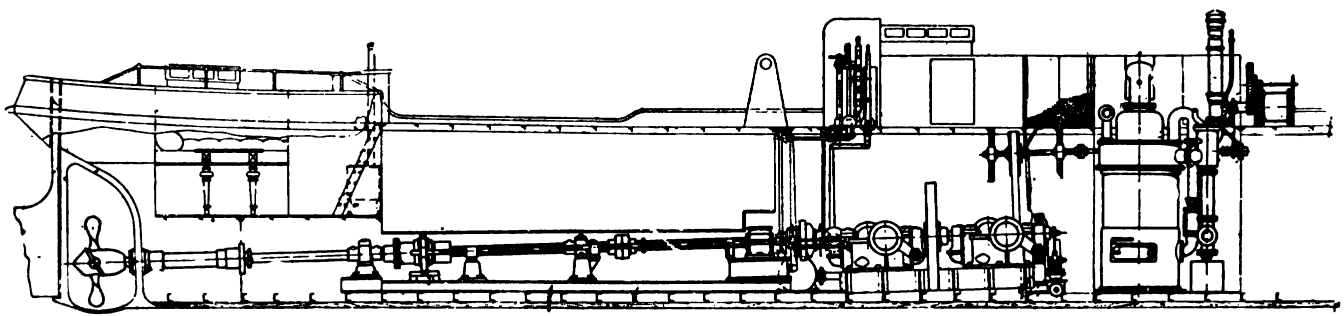


Fig. 36.

boat was the *Lotte*. It has opposite, twin, horizontal gas engines, and one small auxiliary engine for starting.

The *Lotte* is about 131 feet long, 14 feet beam, and of 7 feet draught. The capacity is 240 tons. The horsepower is 100 and the required speed against the current in the Rhine is about 3⅓ knots per hour. Figs. 35 and 36 give a longitudinal section and a plan view.

The producer plant is between the transverse bulkheads, so that no dust nor gases can enter the other parts of the ship. The propeller wheel is 4 feet in diameter. The auxiliary engine runs on gasoline and develops 6 H.P. It drives a blower and the hoists on deck. Besides, it serves as a starting device for the main engine by means of friction-clutch and belt transmission.

In conclusion I give the following two tables for these boats running on anthracite suction gas. One, Table I, runs regularly 150 knots between Saarbrücken and Mühlhausen, and the other, Table II, runs 167 knots between Cologne and Rotterdam. The boat according to Table I has a 16-H.P. producer plant, 240 tons load, runs 12 days with, and 9 days without, load. It makes 11 trips a year, each 150 knots, and the speed is only a little over one knot, average. The coal consumption is taken with 1.21 pounds per H.P. per hour. The other has a 100-H.P. producing plant, 250 tons load, runs 14 days. The average load up and down is 200 tons. It makes 26 round trips a year and 167 knots for each single route. The coal consumption is taken at 1⅓ pounds per H.P. per hour. The trip up the river requires 50 hours, and down 25 hours, or 75 hours for the round trip.

A comparison of the expense in operating a freight steamship and freight suction-gas ship shows about 0.16 cents per gas, ton, knot for the latter, and 0.25 cents for the steam, ton, knot. These costs on steam railroads are about five to six times as high as for suction-gas boats.

TABLE I.

OPERATING COSTS FOR SUCTION-GAS BOAT *LOTTA*.

For wear and tear (expenses per year), 5 per cent.,	$200.00
For wear and tear on machinery, 10 per cent.,	250.00
Interest on installation cost, 5 per cent. of $6,500,	325.00
Fire insurance, 2 per cent. of $6,500,	13.00
Taxes and duties,	150.00
Costs for coal for 11 trips, 16 H.P., 12 days with load, 9 days without load, 12 hours, $5.00 per ton, coal,	122.00
Cost for oil, waste, etc.,	61.00
Wages and salaries,	600.00
Total cost, per year,	$1,721.00

This boat makes $11 \times 150 \times 240 = 396,000$ knots, tons, per year.

$\frac{1721}{3960} = 0.43$ cents for operating expenses per knot, ton.

TABLE II.

OPERATING COSTS FOR SUCTION-GAS BOAT *HALDY*.

For wear and tear on hull, 5 per cent. of $5,000,	$250.00
For wear and tear on machinery, 10 per cent. of $6,250,	625.00
Interests on installation costs, 5 per cent. of $11,250,	562.00
Fire insurance, 2 per cent.,	22.50
Taxes and duties,	975.00
Costs for coal for 26 trips, 100 H.P., 75 hours, 1⅛ pounds of coal at $5 per ton,	582.00
Cost for oil, waste, etc.,	244.00
Wages and salaries,	1,750.00
Total for the year,	5,011.00

This boat makes $26 \times 2 \times 167 \times 200 = 1,736,800$ knots, tons, per year.

$\frac{501.100}{1,736,800} = 0.28$ cents for operating expenses per knot, ton.

I am indebted to the following for much information in preparing this paper:

Truscott Boat Manf. Co., St. Joseph, Michigan; Hercules Gas, Eng. Works, San Francisco, California; Standard Motor Coustruction Co., Jersey City, N. J.; August Mietz, New York; Carl Huber, Berlin, Germany; "Zeitschrift Des Vereines Deutcher Ingenieure," Berlin, Germany; Graf Rudolf Westphalen, Vienna, Austria; Fairbanks Morse Co., Beloit, Wis.

COMPARATIVE TRIALS OF ELEVATING GEARS FOR RAPID-FIRE GUN MOUNTS.

BY JOHN F. MEIGS.

The comparative trials of elevating gears in which the gun was moved in elevation by means of (1) a single hand-wheel and (2) by two hand-wheels on the same shaft, extended from May 1, 1906, to December 1, 1906, and included the firing of 25,000 rounds. Photograph I shows a Bethlehem 6-inch, rapid-fire gun with which the trials were conducted.

THE APPARATUS.

(*a*) The two-hand elevating gear (Photograph I) consists of two hand-wheels on opposite ends of the same shaft, the handles of the wheels being 180 degrees apart, and the pointer using both hands for controlling the gun. The firing trigger (A) is on the handle of the right-hand wheel and is worked by the index finger of the right hand without releasing the handle.

(*b*) The two-hand elevating gear when driven by the left-hand wheel only, constitutes the usual form of elevating gear. In this case the firing was done by means of the ordinary pistol grip (B). A second pointer pointed the gun in azimuth from the right-hand side in both cases.

(*c*) Three sets of speed gears were used by each pointer at each range, and for one complete turn of the elevating hand-wheel moved the gun as follows:

No. 1 Gear 30 minutes of arc.
No. 2 Gear 50 minutes of arc.
No. 3 Gear 75 minutes of arc.

(*d*) A sub-caliber rifle, shown in Photographs III and IV, was rigidly attached to the muzzle of the gun and fired electrically by the firing gear of the big gun.

(*e*) The targets used represented a target twenty-six feet high at the different ranges, and were actually 1.37 inches high for 1,500-yard range; .68 inch high for the 3,000-yard range; .34 inch high for the 6,000-yard range, and .23 inch high for the 9,000-yard range.

The target was moved to simulate the continuous rolling of a ship from 1 to 10 degrees (Photographs III and IV) by means of a cord, one end of which was fastened to the target, and the other end, after passing over a pulley, was moved continuously over the semi-circumferences shown in Photograph II.

(*f*) The curve by which the target was moved up and down simulating the rolling of a ship (Photograph II) is made up of semi-circumferences joined together, and the radii of these are of such lengths as to cause the target, and consequently the gun, to move through from 0 degrees to 1 to 10 degrees depression or elevation respectively, and back to 0 degrees again. The time allowed for this (7 seconds) was the same for each roll, and corresponded to a 14-seconds period for the complete oscillation of a ship. The semi-circumferences were sub-divided into two-second periods, and a time-bell enabled the operator to pass each mark at the proper time.

GUN POINTERS.

Eight gun pointers were in competition throughout the trials, and while the final scores differed slightly, the relative percentages given by the table were practically obtained by each pointer.

CONDITIONS.

(*a*) Each pointer was allowed a trial shot before each string for adjusting the sight. The distance from the sight telescope to the target was 20 feet.

(*b*) Each pointer fired for five minutes continuously with a six-seconds loading interval after each shot.

TABLE OF RECORDS.

The total time of firing as between the records for one-hand and two-hands are equal. For example, the time of firing with one and two hands at the 1,500-yard target and on the rolls 10 degrees to 8 degrees are the same.

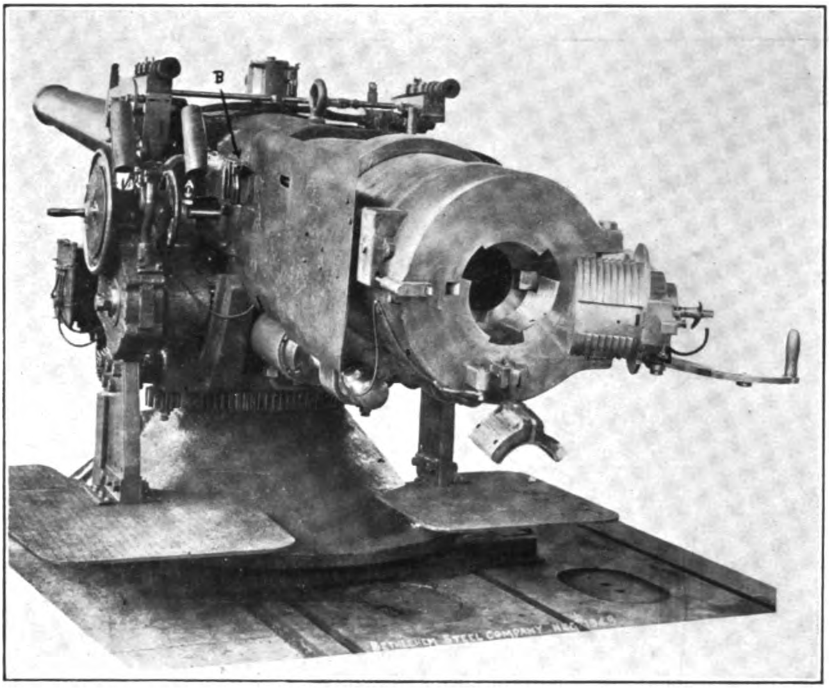
....	Range 1,500 yards.			Range 3,000 yards.			Range 6,000 yards.			Range 9,000 yards.		
	Groups of rolls.			Groups of rolls.			Groups of rolls.			Groups of rolls.		
	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°	10° 9° 8°	7° 6° 5° 4°	3° 2° 1½° 1°
Total shots, 1 hand..	217	356	580	442	1328	1702	275	1055	1377	240	856	1481
2 hands.	242	403	642	518	1544	1837	332	1232	1467	290	1042	1589
Total hits, 1 hand....	153	285	518	225	777	1376	81	442	803	61	286	757
2 hands...	188	345	582	334	1055	1580	128	553	924	101	437	969
Gain in speed of hitting, 2 hands over 1, in per cents......	22.9	21.1	12.3	48.4	35.8	14.9	58.0	25.1	15.1	66.6	52.8	28.0
Gain in accuracy, 2 hands over 1, in per cents.............	10.2	6.9	1.6	26.7	16.8	6.4	30.5	2.5	5.2	37.0	25.5	19.4

NOTES ON TABLE.

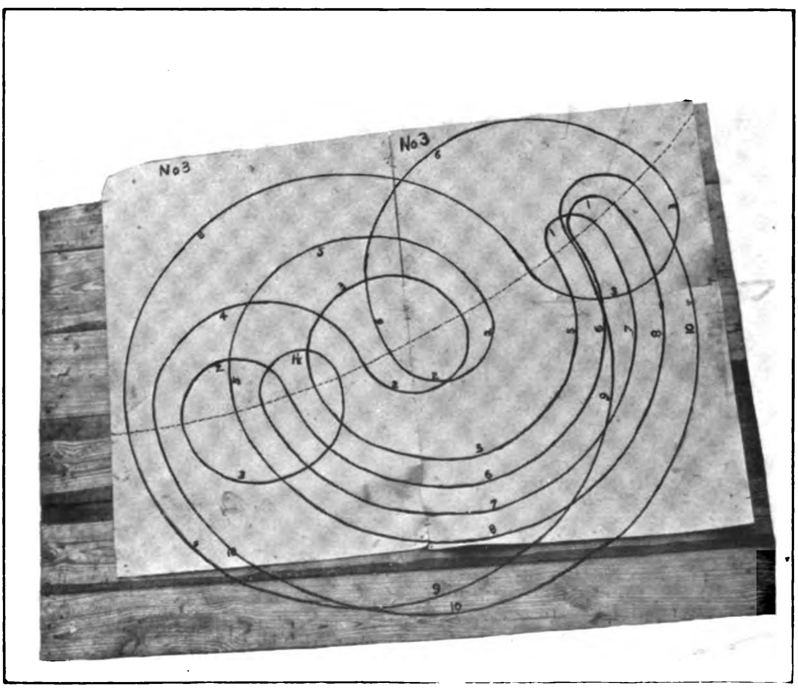
For an actual example, which might occur in practice, we take the figures above at 3,000 yards range and with the ship rolling from 8 degrees to 10 degrees. Imagine two 6-inch guns side by side, one with a one-handed and the other with a two-handed drive. Suppose these guns fire for the same time, with pointers of the same skill, and at the same target. If 100 represents the number of hits made with the one-handed gun, then 148.4 will represent the hits with the two-handed one. Also, under the same circumstances of firing, the percentages of accuracy show that in expending 100 rounds from a magazine, firing at the same speed as above, we shall get 50.9 hits with the one-handed gun, and 64.5 with the two-handed gun. The other shots will be misses. This latter statement is equivalent to this: when firing at the highest speed convenient to pointers, the expectation of getting a hit with any one shot, in the circumstances defined is, with the one-handed gun $\frac{225}{442}$, and with the two-handed gun $\frac{334}{518}$. These fractions are the same as .509 and .645.

Also, the figures show an increased rate of hitting with two hands (an increase in the number of hits per minute) under all circumstances from the greatest to the least difficult. When firing at the nearest target (1,500 yards) and on the rolls 1 degree to 3 degrees, the increase in the rate of hitting is 12.3 per cent., or for every 100 hits in a given time with the one-handed drive there will be 112.3 hits with the two-handed drive. When firing at the most distant target (9,000 yards) and on the rolls 8 degrees to 10 degrees, the increase in the rate of hitting is 66.6 per cent., or for every 100 hits in a given time with the one-handed drive there will be 166.6 hits with the two-handed drive. As stated, in all cases where the difficulty of the practice is intermediate between these two extremes, the two-handed drive shows more hits per minute than the one-handed drive.

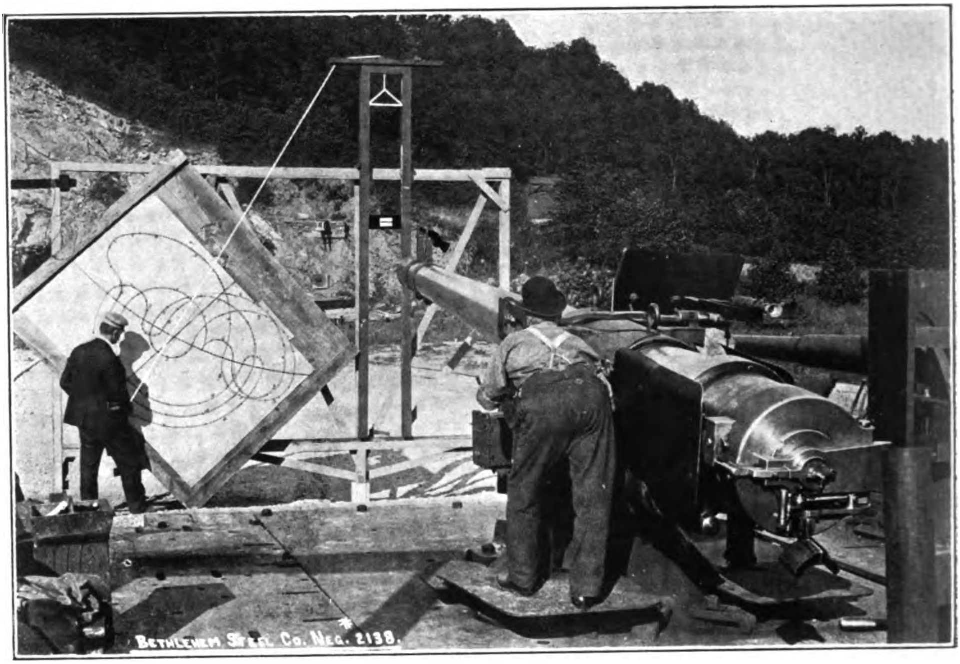
The accuracy of the practice, independently of the firing, is greater in all cases with the two-handed gear. Taking the



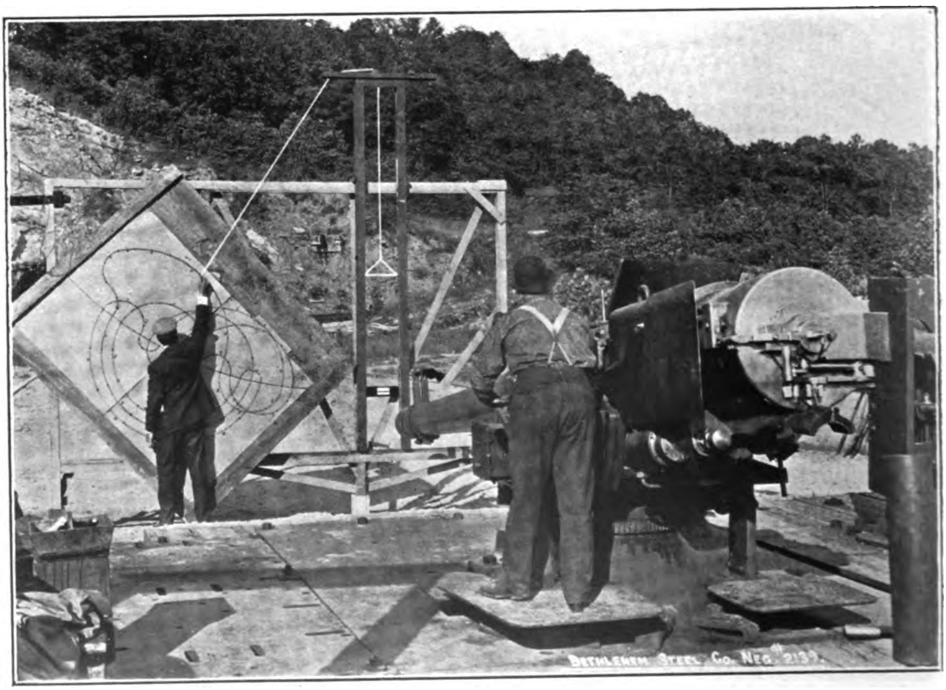
Photograph I.



Photograph II.



Photograph III.



Photograph IV.

two extremes of difficulty as before, the record shows that with the 1,500-yard range and the rolls 1 degree to 3 degrees, the expectation of hitting with any one shot is, with the one-handed gear, $\frac{518}{580}$, or .893, and with the two-handed gear, $\frac{582}{642}$, or .907. With the 9,000-yard range and rolls 8 degrees to 10 degrees, the expectation of hitting with any one shot is, with the one-handed gear, $\frac{61}{240}$, or .254, and with the two-handed gear, $\frac{101}{290}$, or .348. In considering these last figures it must be remembered that they show only the increase in the probability of hitting with any one shot. The increase in the rate of hitting per minute is greater than these figures, and is as already given above. This is because gun pointers will always fire more shots per minute with the two-handed drive than with the one-handed drive. They get their sight on the target oftener. For example, taking again the circumstances of least and greatest difficulty, the record shows that with the 1,500-yard range and in rolls 1 degree to 3 degrees, the number of shots fired with the two-handed gear will be as 642 to 580, as compared with the one-handed gear; or, the two-hand gear gains in speed of firing, $\frac{62}{580}$, or .11. At the 9,000-yard range, and with the rolls 8 degrees to 10 degrees, the number of shots fired with the two-handed gear will be as 290 to 240, as compared with the one-handed gear; or, the two-handed gear gains in speed of firing, $\frac{50}{240}$, or .21. The table of records shows that there is no interruption in the laws of accuracy and speed here shown.

It is to be noted that the results as to accuracy and speed of firing here shown cannot be reached in actual practice with targets at a distance. As the targets were only four feet from the muzzle of the small gun used, the bullets went straight to them, and all inaccuracies due to imperfections in powder and knowledge of range are eliminated. The records show what might be attained if muzzle velocities were very much greater than they are and were maintained, and measure therefore only the accuracy attainable by pointers with service sight and service arrangements for handling the gun.

GENERAL CONCLUSIONS.

The better results obtained by the two-hand elevating gear as shown by the table are accounted for by the following conditions.

(1) The gun can be worked faster with two hands than with one, and the larger the gun the greater the advantages.

(2) The gun can be kept on the target better on account of the ease with which it may be stopped, started or reversed.

(3) The motion of the gun is smoother on account of the greater power to overcome tight places in the gears and also the absence of a dead center with the two-hand drive.

(4) The motion of the gun pointer's body is less with the two-hand drive, and the eye can be kept constantly at the telescope.

DIFFICULTIES EXPERIENCED WITH THE MAIN ENGINE BEARINGS ON SOME OF OUR LATEST VESSELS.

By Lieutenant A. F. H. Yates, U. S. N., Member.

Owing to the wide amount of attention attracted of late by many instances of failures of the main engine bearings of our new vessels, the writer has collected certain data in the case of five vessels, which it is hoped will be of interest to the readers of the Journal.

These failures have resulted in much discussion as to the exact cause, it being believed by some that they were due to the quality of the anti-attrition metal used, and others attributing them to the use of a poor quality of lubricating oil. For this reason special effort has been made to bring out all matters in these connections.

To begin with, the anti-attrition metal required for installation by the builders in all of these vessels is known as the Bureau of Steam Engineering Standard, its composition being as follows:

Best refined copper, per cent., ...	3.7
Banca tin, per cent., ...	88.8
Regulus of antimony, per cent., ...	7.5

by weight, to be well fluxed with borax and rosin in mixing.

To avoid the advertisement of the various oils used, the brands are not mentioned, but it should be remembered that during all of the official trials of these five ships, previous to commissioning, only a well-proved brand of mineral lubricating oil was used, or else lard oil. It was only after commissioning that the so-called specification oils were used.

"*BATTLESHIP A.*"

Preliminary Trial.—During the four-hour run of the preliminary trial of this vessel the main engines worked well except for a hot valve-rod guide on the starboard engine, necessitating slowing down, and the port intermediate guide, which ran warm continually. The trial was successful and the engines were finally reported as entirely satisfactory. A special marine-engine oil was used by the contractors.

The port H.P. crank pin ran hot at a later date, while making 115 revolutions on the measured mile at Rockland, Me., due to a poor quality of lubricating oil used.

Final Trial.—The four-hours' full-power, forced-draft trial was abandoned after the vessel had run an hour and three-quarters, on account of the heating of bearings No. 1 and No. 2 of the port crank shaft and the running of white metal in same. The port crank shaft worked forward from the thrust block until the forward webs of the port forward L.P. and port H.P. and port I.P. crank shafts took on the after fillet of main bearings No. 1, No. 2 and No. 4, melting the babbit metal on the fillets of these brasses. The starboard H.P. crank-pin brasses, the guides of the port L.P. crosshead, and those of the port after L.P. valves also ran hot, although water and oil were used profusely on them. The hot bearings were cooled and the engines ran smoothly for about one hour at the high speeds, after which time the bearings began to run warm, and, although oil was profusely used on them, it was not possible to keep them from heating. Even those bearings which had been very carefully adjusted did not seem to run more satisfactorily than the others.

Owing to the heating of so many bearings, the running of the white metal in some, and the heavy pounding in the starboard H.P. crank-pin brasses, caused by their having been overheated and flattened, it was decided not to run the engines at their maximum speed, and they were therefore run at a moderate speed of approximately 85 revolutions (about 13 knots). The cause of this failure was the use of a poor grade of oil (specification oil).

This trial having been unsatisfactory, it was recommended that *Battleship A* be given another final acceptance trial as soon as a proper adjustment of brasses could be made, using the same high grade of lubricating oil that was used successfully by the contractors on the preliminary trial. The oil used by the builders on the preliminary trial was the same brand as was used by the builders of a sister ship on her trial, which passed off most successfully without the use of any water on the bearings.

It was further recommended that steps be taken to locate the cause of the trouble experienced with *Battleship A's* bearings, whether due to faulty design, faulty babbit metal, or poor quality of lubricating oil.

The vessel returned to a navy yard to have the necessary adjustments made, and during her stay, an investigation was made. Upon examination of main bearings No. 1, No. 2, No. 4 and No. 5 of the port engine, it was found that the after ends of the babbitting of each had softened and pulled off as a result of heating. It was decided to trim out the ends and freeze on other babbit metal. The brasses were then refitted to the journals and the fillets rounded off. The forward and the after L.P. crank-pin brasses of the port engine were found to have too limited clearance (a scant $\frac{1}{16}$ inch of the forward L.P. and a full $\frac{1}{32}$ inch of the after L.P.), at the forward ends. This clearance was increased by cutting off $\frac{1}{16}$ inch from the forward face of the brasses in each case. An examination of the starboard crank-pin brasses showed that the babbit metal had dragged and had completely filled up the oil holes and oil grooves. The metal had also burned. The ship being under hurry orders to be ready for her trial runs, the spare brasses were fitted and the ones removed were re-babbitted. Upon removing the crank-pin brasses of the port H.P., the babbit metal was found to have dragged to some extent and to have partly filled the oil ways, as with the starboard H.P. brasses. There were also cracks in the white metal. The urgency attending the getting away of the vessel prevented anything more being done than to refit the brasses,

but it was recommended that these brasses be re-babbitted and refitted when the vessel should return from sea. Prior to the vessel going to the yard, the port thrust bearing had been set forward $\frac{1}{16}$ inch to increase the crank brasses' clearances, and it was decided to readjust them to the original position, in order that the crank shaft could be moved aft $\frac{1}{16}$ inch.

Repairs having been completed, the vessel proceeded to sea for her trials.

Three hours after the four-hours', full-power, forced-draft trial was started, the port main thrust collars, forward end, ran warm, but were kept under control. All machinery worked well until one-half hour later, when the forward end of the port thrust warmed up again. The forward H.P. crank web began striking on the after end of the cap to No. 2 main bearing on three of the steel parts of the fillet, emitting sparks at each of these places, and at the same time the port H.P. crank pin began warming up on the inboard side; the outboard ahead side had already heated, as was afterwards shown. Water was put on each of these places and an effort made to keep them cool until the end of the run. Several minutes later the babbit in the port H.P. crank pin ran, and, the resultant knock in that bearing being dangerous, the engine was stopped as quickly as possible, and the trial was stopped. The starboard engine was slowed to 40 revolutions and the port engine and its auxiliaries secured and the jacking gear thrown in. The work of taking the port H.P. crank pin and No. 2 port main bearing apart was promptly undertaken. The main bearing was found to be in excellent shape, and the after end was filed smooth where it had taken on the web of the crank, and replaced at a 28 B.W.G. lead. The port H.P. spare brasses were scraped in and fitted as well as possible, though the forward end of the brasses came fairly hard up against the forward web of the crank. The brasses were adjusted to a 26 B.W.G. lead on the bottom and a 28 B.W.G. on the sides. This work was finished the next day, and the engines were warmed up and turned over, and an hour later run up to a speed of 105 revolutions, when the port H.P.

crank pin began running warm again and the trial was given up. An investigation of the port H.P. crank pin and brasses showed that the babbit on the outboard side where it was very thin had worn away until the pin had come in contact with the brass, the brass and pin had expanded, the pin was cut, and the forward edge of the web took on the fillet of No. 2 main bearing cap.

The port H.P. crank pin had been refitted to a 28 B.W.G. lead just prior to the trial, but, as the brass had not been properly centered between the bolt holes when roughed out prior to babbitting, the babbit was too heavy on one side and too light on the other. There was also a slight crack in the babbit in the bottom near the crown, and as the time was too short to re-babbit the brasses it was decided that the brasses would run the trial satisfactorily, but that they should be re-babbitted upon its conclusion.

Several bearings that heated on the previous trial, when using specification oil, ran cool during this second trial, when using a selected oil. This would indicate that the quality of oil was in some measure responsible for the heating of the bearings during the previous trial.

Finally, although the brasses had been carefully adjusted at the yard, the vessel failed to complete her second, four-hours', full-power, forced-draft, final trial, owing to the melting of babbit metal in the port H.P. crank-pin brasses. In view of the fact that a first-class grade of lubricating oil was used throughout this second trial, it was concluded that the failure of this trial was due to the babbitting and partly to improper adjustment. The crank-pin and main bearing brasses were undoubtedly not properly bored or babbitted in places, especially the starboard H.P., port H.P., and port forward L.P. crankpins. Their spares were in the same condition.

The following are some other conclusions that were expressed in connection with this failure:

(a) The oil was good and did its work well, as shown by the condition of No. 2 main bearing after the trial.

(b) The oiling service is inadequate for steaming at full

power. An efficient centrifugal oiler is necessary for the crank-pin brasses, and larger oil pipes and automatic oil gear should be installed. The oil gear for the eccentrics could be much simplified by telescopic oil gear.

(c) The centrifugal oiler suggested should be fitted so that the bearings could be flooded with either water service or from buckets filled with soap and water or oil. As now arranged, when the oil grooves are filled and the bearing begins to pound or run warm, there is no way of getting either oil or water to the heated part. The water service merely drops water on the ends of the bearing, and very little if any penetrates to the center of it.

(d) More clearance should be left between crank-pin brasses and crank webs, and between crank webs and main bearing brasses on the forward end of the web.

(e) Greater margin of bearing surface should be given on all crosshead, crank-pin and main bearings.

(f) The design of the brass, with all the babbit in one slab, is poor, as any knock or heating causes the metal to flow in the direction of least resistance. On several occasions metal had to be chipped off the ends of the brasses and out of the oil grooves, when bearings heated while running at slow speeds, bearings set to a 30 B.W.G. lead.

It might be remarked here that, owing to the limited amount of time available for repairs, none of the changes last mentioned were made, though a change in the oiling gear for the eccentrics, to arrange the run of the oil piping with a continuous descent from cup to strap, instead of running for a considerable distance in a horizontal direction, was authorized.

The fault in making the brasses and in babbitting consisted in the axis center between bolt holes in brass castings not coinciding with the center line of the cored-out hollow for babbitting, causing the babbit to be heavy on one side and thin on the other, at one end, and just the reverse on the other. The holding pieces on the brass casting should have been recessed also, instead of filletted, as they were in places,

so that the babbit, when being hammered in, would hold instead of riding up over those pieces and becoming loose.

"*BATTLESHIP B.*"

Preliminary Trial.—The working of both the main and the auxiliary machinery during the preliminary official trial of this vessel was most satisfactory. As usual on preliminary trials, a large amount of lubricating oil was used, about 540 gallons being expended during the four-hours', full-power, forced-draft trial. A special marine-engine oil was used throughout the trial.

A chemical analysis of the "specification" oil used on this vessel was made at a navy yard several months after she had been placed in service, as it had not given entire satisfaction. Two samples were furnished, one from the top of a tank in starboard engine room and one from the bottom of the tank.

RESULTS:

	Top of Tank.	Bottom of Tank
Specific gravity at 60 degrees Fahrenheit	.9237	.924
Flashing point, degrees Fahrenheit	407.	405.
Free fatty acid, as oleic, per cent	.70	.70
Viscosity (Engler) at 90 degrees Fahrenheit	705.	707.
Viscosity (Engler) at 150 degrees Fahrenheit	163.	164.

These two samples were said to have been taken from a a tank which had remained intact since first filled about two months previous.

The tests show them to be practically identical, no separation of the ingredient oils having taken place.

With these samples were submitted small residues said to have been obtained by straining one gallon of each of the above oils. That from the top oil, amounting to about 1 part in 30,000 of the oil, consisted of siliceous and scaly dirt, evidently from the tank, which had not been thoroughly cleaned. That from the bottom oil, amounting to about 2 parts in 30,000, contained a small amount of dirt, but consisted mainly of metallic oleates (principally of zinc) formed by the action of the fatty acid on the inside coating of the tank. Traces of iron and copper were also present.

In addition to the above samples a sample from a tank in the port engine room, the greater part of the contents of which had been drawn off and strained, was compared with a sample of the strained oil. These were found to be identical in specific gravity and viscosity. The residual oil contained a small amount of the soft, white zinc oleate mentioned above.

Another residue obtained by straining consisted mainly of tobacco, scrap and cotton waste.

The senior engineer officer stated that the oil flowed freely and showed no gumming properties, these results confirming the results of previous tests made in the laboratory when the oil was first received at the Yard. The samples taken for test at that time were perfectly clean and free from dirt.

Final Trial.—A speed trial was made under full power, natural draft, as a portion of the acceptance trial of the ship. The engines were run at an average speed of 105 revolutions for eight hours, at which speed the crank-pin brasses of both I.P. engines pounded siightly and the crosshead brasses of both H.P. engines pounded badly—so badly that it was deemed advisable to heave to for three hours to adjust them before proceeding with the remaining trials at higher speeds. On the conclusion of this eight-hour run a twenty-four-hour run, at 75 per cent. of the I.H.P. developed on preliminary full-speed trial, was begun. The engines were run at about 112 revolutions for three hours. The thump in the I.P. crank pins previously noted grew steadily worse, and at the end of the third hour the thump in the I.P. starboard crank pin became so heavy that the brass heated from percussion and the white metal melted and ran out of the bearing; the trial was, perforce, discontinued and the ship proceeded into port at 11 knots. Before proceeding to the Navy Yard a spare set of I.P. crank-pin brasses, furnished by the builders, was fitted to the starboard I.P. pin; the old brasses were found to be in such condition that it was not deemed safe to try to run up the harbor with them; the soft-metal lining of the brasses was mashed and pounded down badly. The crank pin was scored and rough; it was smoothed before the brasses were fitted.

On the run to the Navy Yard the new I.P. crank pin brasses ran warm. On examination, after arrival at the Yard, it was found that these new brasses, after only one hour of running, were pounded and mashed down badly, the metal filling all the oil grooves. The conclusion was that the metal was entirely too soft.

All the crank-pin brasses were refitted and all were babbitted, except the after starboard L.P. and the two port L.P. brasses. Nearly all of the brasses were found to be sprung out of shape, the back of the brass, in each case, being high in the center over the crown of the brass; also, the bolt holes in the brasses were not in line. The concensus of opinion was that the brasses were neither strong enough nor thick enough at the crown.

During the first run made by the vessel after leaving the Navy Yard the two H.P. crank pins pounded slightly. The brasses were examined and overhauled, and the metal of the starboard H.P. was found to be mashed down into the oil grooves. The starboard I.P. brass, which had been scraped and fitted to the pin at the Navy Yard, was also examined and was found to bear only on the sides and not at all on the crown, as if the brass had again sprung at the center. These brasses were then scraped and fitted to the pins and set up to No. 29 B.W.G.

On getting under way for a run of 90 miles, both engines were found to run well, the port engine especially. At first there was a slight thump in the starboard H.P. crank pin; this thump steadily increased, and at 4 P. M. was very heavy —so heavy that the overhauling of the brass was deemed absolutely necessary before making a long run at sea. On examination, the metal of the starboard H.P. crank-pin brass was found to be spread and mashed out into the oil grooves, and on the cheek-face of the brass, where, at starting, the ends of the white metal had been flush with the surface of the brass, the white metal had been pounded out until it projected $\frac{1}{64}$ inch beyond the ends of the brass, and this after only 90 miles of running.

45

On a five-day run at sea the starboard H.P. crank pins and crosshead pins pounded heavily; on examination, the metal of the starboard H.P. crosshead brasses was found to be mashed slightly, the oil hole in the cup brasses being covered. The port H.P. and port I.P. crank-pin brasses were examined the next day and were found to be brass-bound, while the oil grooves were partly filled with mashed out metal.

This led to a recommendation that the character and qualities of the white metal installed be investigated, especially with reference to its ability to stand the heavy work imposed upon it at high speeds; also that investigation be made to determine whether or not the crank-pin brasses were sufficiently thick through the crown to maintain their shape under moderate pounding. It was ascertained that the same mixture of babbit metal was used for the bearings of *Battleship B* as was used for four other ships by the same builders. This metal was received by the builders from the sub-contractor in a lot, and was similar to the mixture employed by the builders.

In the case of a number of the crank-pin brasses, some of which were re-babbitted at a navy yard, it was noticed that the metal mashed out after a short while at the ends of the brasses so that the white metal projected beyond the surface of the fillet at the ends of the brasses.

"*ARMORED CRUISER C.*"

Preliminary Trial.—About one hour and a-half after the four-hour, full-power, forced-draft trial had started, the starboard engine was slowed down and almost stopped, on account of a knock in the forward L.P. crank-pin brasses. An examination showed everything apparently all right, but when the engine was started up again a knock was heard, and the engine was stopped immediately. The inboard distance piece was found lying in the crank pit, the dowel pins having sheared. During the second four-hour run the fourth main bearing of the starboard engine ran hot, when a full force of water was turned on for half an hour and the bearing was

slacked off. An oiler subsequently found a piece of metal in this bearing. After this no trouble occurred. The contractors used a high grade of lard oil on this trial.

Several months after the preliminary trial was held it was reported necessary to examine and adjust all crosshead, crank-pin and main bearings, most of them appearing to need re-babbitting. All of these bearings on both H.P. and I.P. engines gave trouble continually during a run of about 4,500 miles, every available minute in port during the trip being employed by the ship's force in overhauling them. The port I.P. crank-pin bearing was replaced with a spare one. It first ran warm, and at the end of the return run was knocking badly. The metal appeared to be too soft, having dragged around in flakes, completely filling the oil grooves and accumulating in bullets and rolls inside of the distance pieces. The three crank-pin brasses examined showed the white metal drawn down from the edges of the brasses more than half way around. The lower brasses of the main bearings were not examined but were judged to be in as bad condition as the caps. Owing to these conditions a board of officers made an examination of the bearings. All the bearings of the main engine that were open to inspection were examined, and the white metal was found to be in such bad condition that it was reported necessary to re-line the crank-pin and crosshead brasses, examine when practicable the go-ahead eccentrics and refit the main-bearing brasses throughout. The bearing metal showed such excessive wear, and in many cases such a loose condition, as to make the above work necessary to place the machinery in efficient condition for service. The metal surfaces were found to be dragged in irregular patches, completely obliterating the oil grooves, and the remaining metal in such condition that re-boring would provide too small an amount for wear. It was not possible to examine all of the bearings, owing to the magnitude of the work of stripping them, by the force on board, but from those examined it was apparent that all were in equally bad condition. From the appearance of the bearings and their general history it was

concluded that the primary cause was the use of an inferior quality of oil.

While there was no record of the bearings having run excessively hot, it was stated that it was impossible to run them cool, even with an excessive supply of this oil and at comparatively low speeds, without using water service.

A chemical analysis of the babbit metal used in the bearings of *Armored Cruiser C* was made at a navy yard with the following result:

Copper, per cent.,	4.30
Tin, per cent.,	88.05
Antimony, per cent.,	7.53
Lead, etc.,	Traces.

This corresponds closely with the mixture prescribed.

The oil used on *Armored Cruiser C*, as stated above, was specification oil. It was received on board in a lot of 2,000 gallons. This oil had been rejected on first inspection, but was finally passed, and 1,941 gallons of the oil was used. On account of indications of hot bearings orders were issued to be liberal in the use of oil. During the vessel's run of about 4,500 miles, the four H.P. and I.P. crank-pin bearings ran warm five times, and three main bearings once each. As a rule the warm crank-pin bearings followed close upon readjustments.

Sample bottles of the oil taken from the gauge glasses shortly after receipt on board showed from ⅛ inch to ¼ inch fairly solid sediment. Samples taken from a 1,000-gallon tank with only 150 gallons remaining showed about the same. Eleven gallons of sediment were taken from a tank that had contained 1,100 gallons of oil. Of this two gallons were of the consistency of soft soap.

Water was kept constantly on the H.P. and I.P. crank-pin brasses, first to reduce the temperature and afterwards to reduce the knocking. The oil did not produce a lather with either fresh or salt water, and did not appear to cling to the bearings. This was noticeable on the main bearings where

the shaft was nearly dry on the inboard side, even when oil was being poured into the outboard side of the hand hole.

Armored Cruiser C was not in readiness for her final trial on the contract date, the re-babbitting of her bearings not having been finished.

As it was thought possible that the failure of the crank-pin bearings of this vessel was, in part, due to an insufficient supply and bad distribution of oil, two more oil holes in each crank-pin bearing were drilled, near the ends, with tubes and oil boxes located alongside the present ones, using drip-feed boxes, without sight-feed attachments.

Subsequent to the above an examination of samples of the oil used, of the bearings that had been removed, of pieces of solid metal from the bearings, and of specimens flaked off, was made by the builders of the vessel. The analysis of the solid metal was as near as possible that which was originally put in (Government mixture), while that of flaked specimens was very much different, being mostly tin, which melts at a very low temperature. The analysis made showed the following compositions:

	Tin.	Antimony.	Copper.	Lead.
Government standard	88.8	7.5	3.7	...
Port H.P. crank pin	89.57	6.66	3.7	.07
Flake from port H.P. crank pin	94.8	3.95	1.12	.14

This examination would seem to indicate that the metal of the bearing was, in spots, heated to above the melting point of tin (446 degrees), when the tin separated from the other metals and "flaked" or "dragged." The heating might have occurred from bad oil, too snug fit, or from an improperly-fitted bearing; most probably the causes in this case were bad oil and too snug a fit.

"*ARMORED CRUISER D.*"

Preliminary Trial.—The four-hour official trial of *Armored Cruiser D* was accomplished without the least trouble arising to cause a pause for an instant. The main engines ran with remarkably little vibration, none of the main bearings and journals heating, and the water service being used only on the

crosshead guides and thrust bearings. After the trial the following parts examined were found to be in excellent condition: after L.P. and H.P. crank pins starboard engine, after L.P. crank pin port engine, and top brass of after L.P. crankshaft bearing on each engine. The crank pin of port I.P. was slightly scored. The contractors had both high-grade lard oil and a selected grade of engine oil on board for lubrication purposes.

Upon the return of the vessel from a trip of about 4,500 miles, the bearings of the main engines were reported in bad condition. On the trip out the engines worked without much noise, but on the return trip the bearings became looser, and the knocking, especially in the high and intermediate on both sides, increased to an excessive degree. The starboard I.P. crank brass and port I.P. slipper were taken down and the babbit found to be in very bad condition, the oilways being almost obliterated. The amount of play in the bearings of the crank pin and guide examined was about ⅛ inch. There was little trouble with heating in these bearings on this trip. The bearings on the L.P. were not examined at this time, as they had given practically no trouble. A later examination showed the bearings, generally, to be in much worse condition than was at first supposed, and for this reason a more complete examination was ordered. Immediate relining of crosshead slippers, go-ahead eccentrics and crank-pin brasses was found to be necessary. The white metal was found to have dragged in all the brasses, and the oil-ways were obliterated. In view of the fact that the babbit metal withstood the forced-draft trials, whereon at least three times the horsepower of that developed on the recent run was obtained, it was concluded that the primary cause was the use of a poor quality of lubricating oil.

The vessel proceeded to a navy yard for the necessary re-lining of her bearings. Upon arrival an examination of the oil in her tank was made. The amount remaining on hand was about 170 gallons, of which 150 gallons was in one tank and about 20 in another. One sample (b) was drawn from the former

and two from the latter. Of the two latter, one (c) was drawn from near the surface through a pet cock on the gauge glass, and the other (d) through the spigot connected to the pipe leading from the bottom of the tank. A sample (a) of the oil that had been used by the contractors on the preliminary trial on hand was also drawn. Immediately after drawing off, a slightly murky sediment was apparent in all samples, the most being in samples (a) and (b), and the least in the samples (c) and (d), but with very little difference. After standing some sixteen hours, samples (a), (c) and (d) showed a cloudy appearance throughout the bottle, while sample (b), specification oil, looked perfectly clear. There was no apparent increase in the amount of sediment in the bottom of the bottles. A comparison of the contractor's special oil with the specification oil, by rubbing oil in the hand with a little water, showed that the former gives a quicker and more copious lather and works cooler in the hand.

Specification oil was used during the vessel's run, with the exception of 200 gallons of the contractor's special oil used on the starboard engine. The only time that really hot bearings were experienced was once with the valve-stem guide of the starboard I.P. engine, due, probably, to a wrong combination of valves on the assistant cylinders, and once on the port H.P. engine, which was caused by insufficient oil. In both cases the heating was quickly reduced, and no further trouble was experienced. No other bearings developed excessive heat, and the oil would at times show a little lather.

The brasses of the port H.P. engine showed that the white metal at the sides of the brass was squeezed, both in the direction of motion and also in the opposite direction, whereas, if their condition was due entirely to bad oil, it should show only on the go-ahead side. The H.P. and I.P. crosshead slippers showed the most lost motion, although they received a liberal supply of oil.

Of the four samples of oil, the three samples of specification oil showed a decided clearing in the upper part of the bottles and a settling in the bottom, the sample of the contractor's

special oil apparently undergoing the same process, but more slowly.

Examination of the top eccentric strap showed an even wear, with no signs of heating and a squeeze of the white metal in both leading and following edges of the segments.

The cap of No. 6 port main bearing showed the white metal in good condition, but there were two cracks about 6 inches long, extending from the hole for tallow box.

A sample of white metal hammered down and broken showed very nearly the same characteristics as Parsons, and suggested the possibility that the metal was not properly hammered and peened into place.

In connection with the failure of the bearings of *Armored Cruiser D* and *Armored Cruiser C*, comparative compressive tests of three anti-friction metals were made at a navy yard. The test pieces of naval white metal were taken from stock. Parsons white brass showed slightly the best flowing qualities, being pressed thinner than the naval metal, yet showing the least breaking around the edges. Magnolia metal showed a distinct lead color and was about 60 per cent. heavier than either of the others. The metal from which test pieces were cut was first poured. The compressibility of the different metals may be compared by noting the final reduction of heights of approximately equal compression loads.

....	Naval white brass.		Parsons white brass		Magnolia metal	
Test piece No.	N–I	N–II	P–I	P–II	M–I	M–II
Original diameter, inches	0.749	0.75	†0.749	†0.749	0.743	0.75
height, inches	0.75	0.75	0.759	0.758	0.751	0.748
Elastic limit (about), lbs.	*8,000	8,000	5,000	5,000	5,000	4,500
Final compress loads, lbs.	36,000	37,000	36,500	36,800	37,000	36,500
Final height, inches	0.2	0.192	0.1805	0.18	0.161	0.166

* Less than. † Full.

An opinion was expressed during the work of re-babbitting these bearings that possibly the brasses contained too great a thickness of white metal.

By a comparison of the metal in various bearings, both in *Armored Cruiser C* and *Armored Cruiser D*, it would seem

that those containing about $\frac{7}{16}$ inch of metal (above the ribs) stood their load much better than those containing more than this amount. This might be partly due to the fact that it is more difficult to properly peen the thicker metal in place. None of the crank-pin brasses of *Armored Cruiser C* were fitted with sufficient or proper anchorages for the white metal at the parting of the boxes. The metal in the bearings of this vessel was, for this reason, torn away from the boxes at this point. Similar conditions were found on *Armored Cruiser D*. Dovetailed anchorages were therefore cut in the boxes.

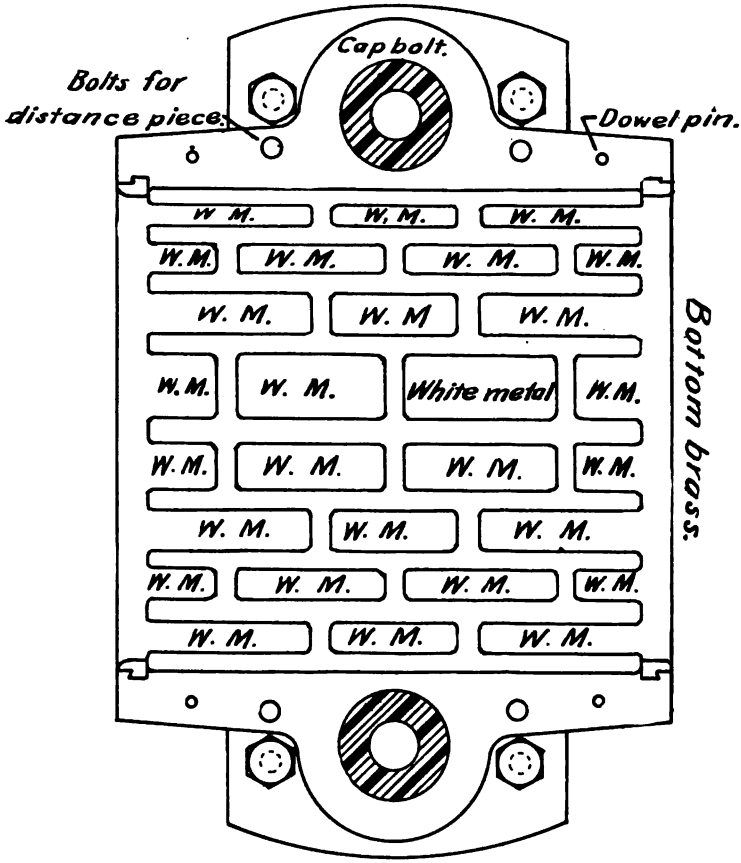
The Board of Inspection and Survey, on final inspection of this vessel, reported that the size of the oil holes in the crank-pin brasses be increased.

Armored Cruiser D was not ready for her final trial, as the work on her bearings remained uncompleted on the date required therefor.

"*ARMORED CRUISER E.*"

Preliminary Trial.—Difficulty was had with the crank-pin bearings only, beginning first in the I.P. bearings, and being worst with the port engine. These bearings were babbitted in the manner shown in sketch No. 1 with white metal composed of 3.7 per cent. best refined copper, 88.8 per cent. of Boustead tin, 7.5 per cent. of antimony, Boustead tin being substituted for Banca tin by special authority of the Bureau.

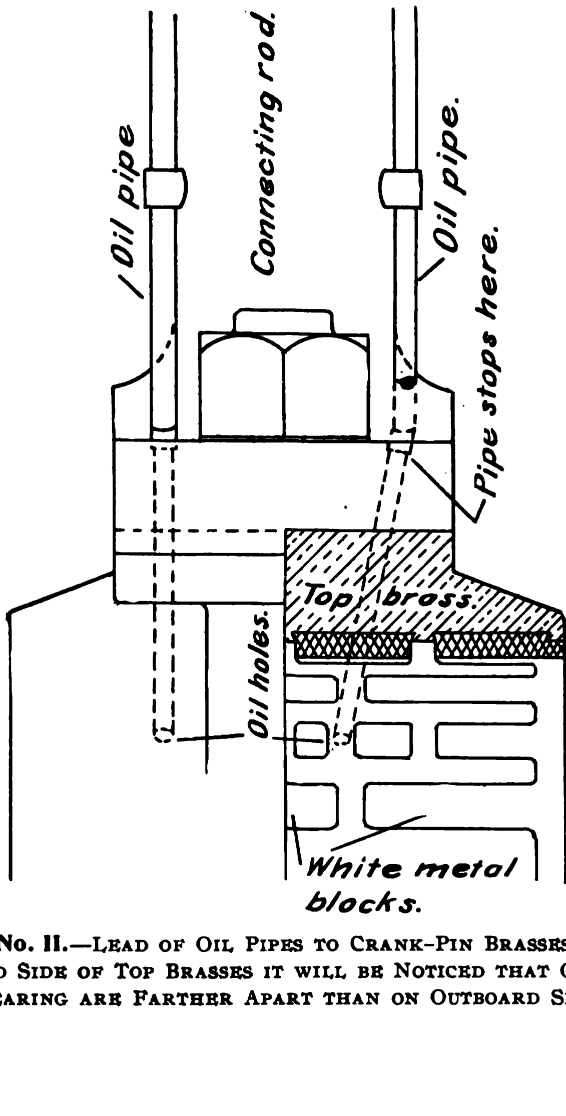
From the sketch it can be seen that the oil channels take up much space, the bearing surface being considerably reduced. On the standardization run, when the engines were speeded up to 130 turns, the I.P. of the port engine developed 5,621 I.H.P., while the H.P. made 2,936, the forward L.P. 2,945, and the after L.P. 2,852. This imposed excessive work on both I.P. engines, and knocking began after the first hour, increasing quickly until most of the metal was pounded out in the port bearing, and the pin was running on the brass, scoring it considerably. All the water possible was kept running on the bearings and there was no sign of overheating,



Sketch No. I.—METHOD OF BABBITTING FOR STANDARDIZATION AND FIRST ATTEMPT AT FOUR-HOUR RUN. METAL POUNDED OUT.

but as soon as the metal began to be pounded down the oil holes in the club end of the connecting rod filled up, and no more oil got into the bearings. The brass plates over the holes through the crank pins were left off, and no grease put in the pin. The oiling gear for the crank-pin bearing consists of four oil tubes led down along the connection rod from the oil cup at top, fed by drip under the cylinder (sketch II). On reaching the club end the oil tube bends out and passes through the club end and projects about ¼ inch

down into the hole in crown brass, the hole being carried on through the brass to the bearing (see sketch No. II.) So on each bearing there are four holes in the upper brass, two inboard about 11¼ inches apart, and two outboard 5¾ inches apart. With the type of babbit originally fitted sufficient oil seemed to feed into the bearing, the trouble being caused by excessive horsepower in the I.P. engines and insufficient bearing surface to stand the added strain. The engines are



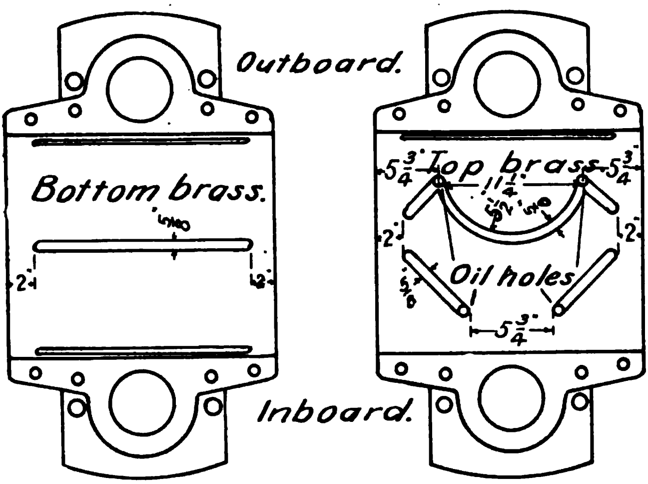
Sketch No. II.—Lead of Oil Pipes to Crank-Pin Brasses. On Inboard Side of Top Brasses it will be Noticed that Oil Holes to Bearing are Farther Apart than on Outboard Side.

designed to have, at moderate speeds, one-third load on H.P., one-third on I.P., and the other third divided between the two L.P. cylinders. When high speeds were reached this did not hold good and the I.P. got most of the work. The crank-pin brasses were fitted to 28 B.W.G. lead.

After this run the spare brasses were fitted on both I.P. pins, and the first attempt was made to run the four-hour trial. In order to lessen the load on the I.P. engine, all L.P. cut-offs were run back to the five-tenths mark on the slot in the arm of the reversing shaft, giving a cut-off in the L.P. of about .55 stroke. After running so for a quarter of an hour they were run back to the eight-tenths mark on the port engine, giving a cut-off of about .35 stroke, and a little later on they were run to seven-tenths on the starboard engine, and these positions were retained until the trial had to be given up. Before the run began the port I.P. crank pin was knocking slightly, and this increased gradually until, after forty-five minutes running, the babbit was all pounded out of the port I.P. bearing, and the pin was again running on the brasses, the pounding being terrific. The starboard I.P. was also knocking, but not so badly. All water possible was kept running on all bearings. Two hours and a half after the trial began the inboard parting piece flew out, wrecking the crank-pit guards, stay rods, platform and water service. Examination showed that the bolts holding it in had broken at the first thread, and that the dowel pins had sheared. The nut on outboard cap bolt had backed one turn and a half, and the set pin was bent up. Practically all the white metal was cut except between the ribs on the brasses. There was no sign of melting of the metal, and the bearings were not hot. In spite of the relief given by altering the cut-off of the L.P. cylinders, the metal had again pounded out. Examination of the brasses showed pounding in the starboard H.P., the starboard forward L.P. and the port forward L.P., but none had heated.

On returning to the ship yard the white metal was melted out of all brasses except the port H.P., and they were re-bab-

bitted with a white metal selected by the builders, supposed to be somewhat harder than the Navy standard composition. In re-babbitting, the bearing was poured practically solid, the oil channels being very much reduced and the bearing surface very much increased (see sketch No. III). This was going to the other extreme and not giving much room for the cir-



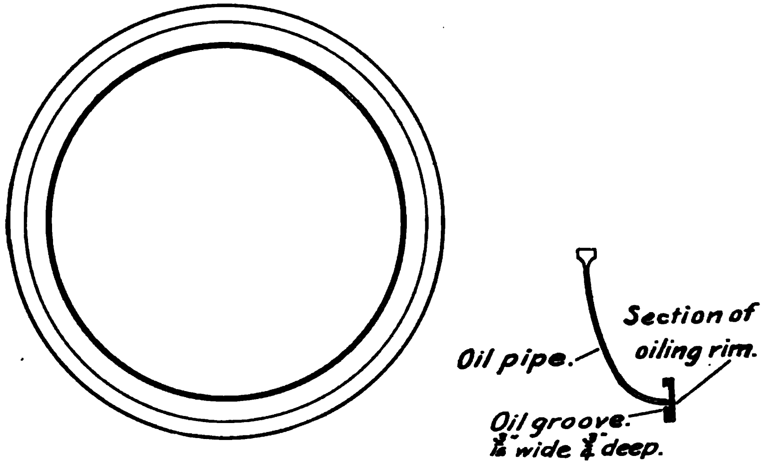
Sketch No. III.—BRASSES AS BABBITTED FOR SECOND, THIRD AND FOURTH ATTEMPTS AT FOUR-HOURS' TRIAL. METAL MELTED OUT UNTIL CENTRIFUGAL OILERS WERE FITTED. OIL CHANNELS $\frac{1}{4}$″ TO $\frac{3}{8}$″ DEEP.

culation of oil in the bearing. The pins were filed down to smooth off the scores, the brasses refitted and the crank-pin holes filled with Albany grease and the brass plates put on. To fill all the pins, 1,120 pounds of this material were necessary. Two one-half inch holes in the pin gave this grease access to the bearing, and centrifugal force forced it out. In addition the regular oiling gear was in use. The L.P. cut-offs were run back to full length of slots in arms of reversing shafts and the brasses were fitted to about 26 lead B.W.G. As soon as the trial began all of the bearings were flooded with water. After running $2\frac{1}{2}$ hours the port forward L.P. crank pin showed evidence of heating, the white metal melting out, and the trial was stopped. The brasses outside were

not so hot that the hand could not be held on them. During this trial the engines worked very smoothly, only a slight knocking being apparent in the port after L.P., and the bearings seemed to be well oiled, from the amount of grease thrown out on the guards. On each revolution the oil seemed to be thrown out and up the oil pipes on the connection rod, against the under side of the cylinder, and with it some of the Albany grease. Examination of the bearings showed the metal in the port forward L.P. melted out to great extent, but the engine was so quickly stopped that there was considerable metal still left. All the Albany grease was out of this pin, and the inside of the pin contained some melted white metal and showed evidence of considerable heat. Very little grease remained in any of the other pins, the greatest amount being 6 pounds. This would indicate that the bearings were all right as regards amount of surface as long as they were lubricated, but so much lubricating material was thrown up out of the oil pipes that very little from that source got into the bearings. With sufficient Albany grease to last the four hours the ship could have made the run, but as 1,120 pounds had been expended in 2½ hours, or at the rate of about 5 tons per day, that system of lubrication was hardly practicable. Still the faults did not seem quite clear, so the vessel was made ready for another attempt. No changes were made in the bearings, though all brasses were gone over, smoothed up, burrs chipped off and the oil holes and channels cleaned out. On the theory that closing up the holes in the crank pins caused the overheating, the brass plates were left off this time and the holes left open, relying only on the regular oiling arrangements. A third attempt was then made at a four-hour trial. On speeding up to the required number of revolutions oil was again thrown out of the oil pipes, and the port forward L.P. crank pin gave evidence of heating. The engines were slowed down and extra water service turned on until the pin was cool. Another attempt was then made, but within about three minutes after the beginning of the trial the port forward L.P. bearing began to get hot, and the trial was called

off. While the ship was returning at slow speed some bath brick and oil were run into the port I.P. bearing to smooth away the scores. Examination of the bearing showed that it had begun to melt, but the engine had been stopped so quickly that very little metal was gone. It was then decided to fit centrifugal oilers, making three systems of lubrication for the crank pins. It would perhaps have been possible to increase, somewhat, the oil channels without sacrificing too much bearing surface, for in all these trials the port H.P. crank pin, fitted with the regular Navy system of bearing blocks made of Navy composition, had never given any trouble whatever, and was in perfect condition.

The centrifugal oilers, as fitted, were composition rims with an open groove in the outer edge, the oil feeding into the groove from an oil pipe led to the oil box on the engine framing. One apparatus was fitted for each crank on the forward side of the web, being secured to it by clamps bolted on the side of the web. When the engine turns over the oil fed into the channel is thrown out toward the rim of the wheel by centrifugal force, and from there led by another oil pipe to a new hole $\frac{3}{4}$ inch diameter bored through the crank pin between the two holes already existing for interior lubrication. As there was very little clearance between the web of the pin as it revolved and the water-service pipes to the main bearings, the oiling rim had to be very thin, and this is its principal fault. At the most it could hold only $\frac{1}{4}$ pint of oil, and might be easily clogged. Then with syringes oil was squirted constantly into the groove during the trials. The brass oiling ring or rim is $3\frac{1}{4}$ inches wide and $\frac{7}{16}$ inch thick. The oil groove in it is $\frac{3}{16}$ inch wide and $\frac{3}{4}$ inch deep. Sketch No. IV shows the details not apparent on sketch VI. After having installed centrifugal oilers the contractors made a run for their own information, keeping the engines at 120 turns for about an hour and a half, everything working well, with the lubrication good and no signs of heating or pounding. The brasses of the intermediate and low-pressure crank pins were removed and overhauled, and found to be in excel-



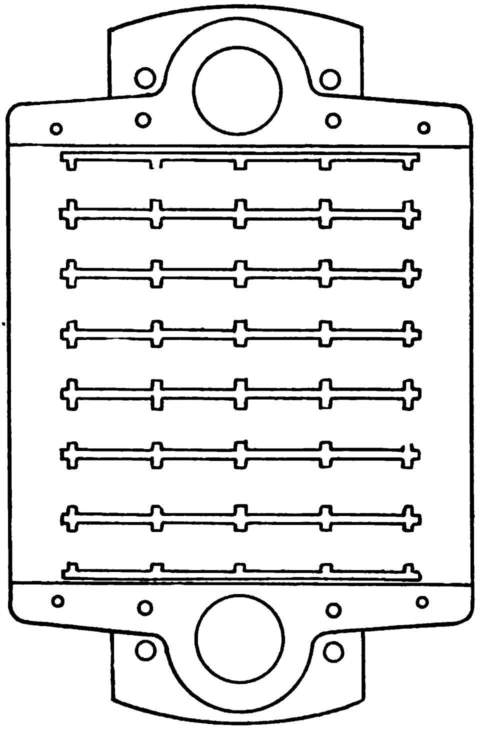
Sketch No. IV.—CENTRIFUGAL OILERS, "CALIFORNIA."

lent condition. In refitting them they were put on very loosely. Leads on the low-pressure crank pins measured from 24 to 27 B.W.G. lead, on the starboard intermediate 22 to 26, and on the port intermediate 21 to 27. In places the leads, which were about 20 B.W.G. originally, showed no contact whatever. The port I.P. crank pin had become so smooth that no traces were left of scoring. The ship was reported ready, and the fourth attempt was made at a four-hour trial. Before the trial began there was a slight knocking in both I.Ps., but this increased only slightly during the trial. Slight knocks developed also in the after L.P. on both sides, but did not increase to any great extent. A great deal of water and oil was run on and into the bearings, and it is probable that the bearings, being continually flooded with both, prevented any increase in the knocking. The ship passed this trial successfully, and the bearings, on examination after the trial, showed little injury. The port I.P. crank pin, which had been scored most on the standardization run and on the first attempt at four-hour run, was worn quite smooth, showing no traces of the former scoring, and was only .003 out of round.

Examination of the pins and brasses showed that greatest stress comes on the bearings during two quadrants of the

revolution, and shows effect in the upper inboard quadrant and the lower outboard quadrant of the bearing.

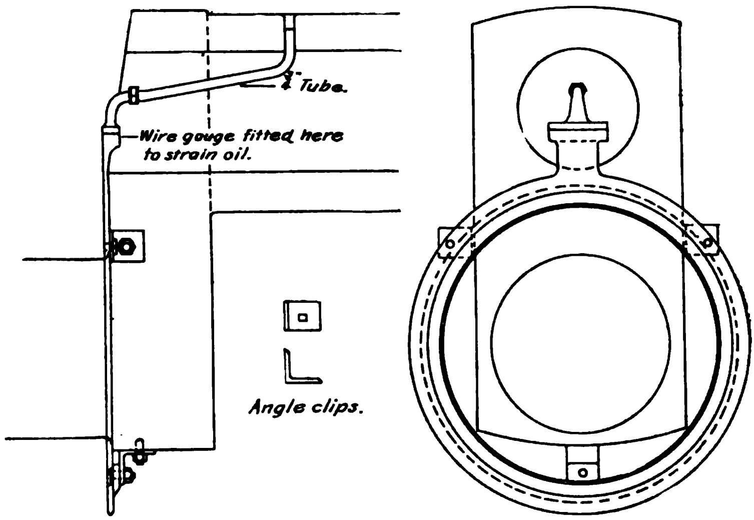
Criticism was passed on the oil grooving of the bearings of this vessel, the opinion being expressed that a bearing such as shown in Sketch V, which has been used with satisfaction



Sketch No. V.—BEARINGS AND OIL CHANNELS OF "NEBRASKA."

in another vessel, would be much more efficient. The opinion was also expressed that the system shown in sketch I would work on all bearings except the I.P., for which a special arrangement would be necessary.

It might be remarked, in conclusion, that the white metal used on the latter trials did not prove superior to the Navy standard metal, as it failed after two attempts of a few minutes each in the third trial for four-hours, but did prove satis-



Sketch No. VI.

factory on the fourth and fifth trials, after the installation of centrifugal oilers. This tended to show faulty lubrication rather than faulty metal used.

To conclude:

On account of the limited amount of space allowed for the installation of machinery in U. S. Naval vessels, the engines must be designed with a minimum amount of bearing surface. This is so to even a more marked degree in our service than in other services, and bears on the question as to whether or not the efficiency, as a whole, of our vessels is increased by the great sacrifice of machinery space for the use of other departments. Under the actual conditions, however, with the high speeds necessary, a high pressure per unit of bearing surface results. Particular care is therefore necessary in designing the bearings and in selecting a suitable material therefor. With perfect design, perfectly even running and perfect lubrication, the kind of material used would to a great extent be immaterial, as, under such conditions the bearing surfaces would be separated constantly by a film of oil, and only fluid friction and the wear of fluid against metal would result. No engine, however, can run absolutely smoothly,

being subject to varying stresses due to thrust, lateral motion and the reversal of the direction of motion, and as a certain amount of grit is ever present, even with the best grades of lubricating oils, the question of the material becomes an important one.

This material must be of such a nature that friction will be reduced to a minimum, and such that it will not heat readily; it must be capable of withstanding the chemical action of the oil, and of oil and salt water; it must be such that it can be easily removed and replaced; it must have a reasonably high melting point, and must stand not only the mean load but the load resultant from intermittent jerks, without deformation; and, finally, it must be such as not to cause undue wear either of the journals or of its own surface.

It seems to be the general practice of marine engineers to use for this purpose one of the alloys known as white metals, meeting, as it does, the necessary requirements of reducing friction to a minimum, of not heating readily, and of being easily removed and replaced. Various metals may be used in the composition of a white metal, the most common of which are tin, copper, antimony, lead and zinc. The alloy selected by the Bureau of Steam Engineering for the main bearings of naval vessels is, as stated at the beginning of this article, made up of copper, tin and antimony. The copper is introduced to give a high melting point, the tin for strength, and the antimony for hardness. The tin and antimony together give the metal stiffness, while the tin and copper tend to minimize chemical action. To give the necessary strength it will be noted that a high percentage of tin is used; the antimony, being more or less brittle, a limited amount is used to supplement the strength by hardness; only enough copper is used to give the desired melting point. Lead is the best wearing metal known, and being very soft and plastic, forms an excellent bearing surface, but, because of the difficulty in overcoming its tendency to flow, it is omitted from the above-mentioned alloy. Zinc adheres to iron when even slightly heated, and becomes brittle under the effect of heat, so is

little used. The standard Navy metal is generally accepted as one of the very hardest alloys that can be classified as a white metal.

The compressive test, such as was made of the metal from the bearings of *Armored Cruiser D*, is of great importance, as it indicates whether the alloy is hard enough to support the required load without deformation, whether it is so brittle that it will split under its load, and whether sufficiently plastic to run cool under the varying stresses to which it is subjected in actual service.

Great care is necessary in the mixing and casting of the metal originally, as well as in the re-melting, re-pouring and peening when the bearings are made. The most expert hands are required for this work as a special practical knowledge is necessary. It is of vital importance that the metal be carefully peened in place in order to compress and harden the metal and fill up all crevices, after which it must be accurately bored and scraped to a proper fit on the journal.

There is probably little difference in the value of the generally accepted metals by reason of their composition, as the good or bad bearing undoubtedly depends more upon the re-melting, re-pouring and peening than on the metal itself, always assuming that the bearing is properly adjusted and the lubricant good.

In the design of the bearing there must be considered—the construction of the bearing, whether the white metal should be in one slab or more, and of what thickness, the clearance to be allowed, the arrangement and extent of oil grooves, the means of supplying the oil, and the point of admittance of the oil; of late, the question of the necessity of centrifugal oilers has come up.

From the above conclusion it can be seen that there are a multitude of points bearing upon the efficient performance of main engine bearings in service. Faults of one nature may give rise to difficulties at high speeds, whereas others may even cause failures at the lower speeds. It is a matter of great difficulty, in cases such as are cited in this article, to

pin down the cause to but one of the many reasons, for, to sum up once again some of the causes for hot bearings:

(1) Insufficient area of bearing surface;

(2) Strength of the casing;

(3) Wearing surfaces out of line or badly fitted;

(4) Improper adjustment;

(5) Improper or insufficient lubrication, due to poor means of supply, poor oil, presence of dirt or grit, or to neglect;

(6) Unsuitable bearing metal.

It will be seen that the design, the adjustment, and the management are all three jointly concerned.

It has not been the object of the writer to put forth herein his reasons for the causes of the failures cited in the cases of the five vessels mentioned, but to present the facts of each case before the readers, that they may draw their own conclusions.

It seems only just to close these remarks with the statement that our engines are required in many instances to develop a much greater horsepower than the specifications of the vessel have called for, in order to attain the speed required by the contract. The uncertainty due to such conditions, coupled with the fact that so limited an amount of space is allowed for the installation of the machinery, requires unusually careful judgment in the design of our bearings.

Some time has elapsed since the writer began this article, and in the meantime the various defects noted have been remedied by re-adjustments and the use of high grades of lubricating oil. In consequence thereof there have been no adverse reports on the working of the bearings.

U. S. REVENUE CUTTER *ITASCA*.

THE REHABILITATION OF THE U. S. S. *BANCROFT*.

By First Assistant Engineer, Charles S. Root,
U. S. R. C. S., Member.

A steam vessel having become desirable for use in training cadets of the line and engineer corps of the Revenue Cutter Service, to replace the bark *Chase*, the U. S. S. *Bancroft* was provisionally transferred from the Navy to the former service on January 20, 1906. On June 30th of the same year the transfer was confirmed by act of Congress. After the final transfer the vessel was rechristened *Itasca*.

For the information of those unfamiliar with the vessel the following description is given.

Intended for a practice ship for the Naval Academy, the *Bancroft* was designed to contain everything in the shape of machinery and equipment that was usually found on a large protected cruiser, and this scheme was followed out even to the extent of fitting steam turning gear to her diminutive propelling engines.

She was built by the Samuel L. Moore and Sons Co., of Elizabeth, N. J., and cost when fully equipped and ready for sea, $431,281.63. Her rig was that of a barkentine to topgallant sails without a bowsprit. She was built with poop and topgallant forecastle, ram bow, protected rudder and steering gear.

A light protective deck extended from stem to stern, with a cofferdam for cellulose from forward of the forward magazine to aft of the after magazine, including the machinery and boiler spaces. No double bottom was provided, but all other watertight sub-divisions were fitted that were usually found in vessels of the *Baltimore* class.

The armament consisted of four 4-inch R.F. rifles, two 6-pounder Hotchkiss, two 3-pounder Driggs-Schroeder, one 1-pounder Hotchkiss, one 37-mm. Hotchkiss, one Gatling gun and two torpedo tubes discharging above water.

She was launched April 30, 1892, and the official four-hour trial was held January 26, 1893.

MACHINERY.

The boilers, two in number, were of the gunboat type. They were placed side by side in one watertight compartment and were fired from the after end. The uptakes discharged into one funnel. Forced draft was installed on the Kafer closed ash-pit system. The main feed pump was in this compartment.

The engines, two in number, were the usual three-cylinder, inverted, direct-acting type, with the H.P. cylinders forward and the L.P. cylinders aft. The H.P. and M.P. cylinders were fitted with piston valves and the L.P. with an ordinary double-ported slide valve. All valves were worked by Stephenson link motion with double-bar links. The bedplates were cast in three sections. The bedplates, pistons, cross-heads, cylinder- and valve-chest covers were of cast steel. The cylinders were supported on wrought-steel columns with steel tie rods.

Steam-hydraulic reversing gear was provided. The crank shaft was in three interchangeable sections. The shafting was all of mild steel and hollow.

The propellers, of manganese-bronze, were built up, three-bladed true screws and turned outboard.

The main condenser was cylindrical with a composition shell, set abaft the propelling engines and in the center of the vessel, the tubes being athwartship. A small auxiliary condenser with air and circulating pumps was installed, for use when not under way.

An air pump of the twin-cylinder type was fitted, and also a centrifugal circulating pump for the main condenser, together with the usual naval outfit of auxiliaries.

In the original plans of this vessel there were nine bunkers below the protective deck, one on each side of the forward magazine, one cross bunker forward of the boilers, two bunkers on each side of them and one on each side of the engine room. In addition, there were eight bunkers between the protective and berth decks so arranged as to trim into the lower bunkers. The capacity of the bunkers proper was about 115 tons.

After the transfer of the vessel to the Revenue Cutter Service, on February 8, 1906, a board of survey was appointed, with orders to recommend such repairs as would be necessary to put the old steam machinery of the vessel in efficient condition for sea service. The report of the board showed that most of the machinery—with the exception of the propelling engines—was practically worn out and not worth repairing. Upon this report the Department decided that it would be necessary to install practically new machinery throughout, with boilers of a different type. Extensive repairs to the hull were also planned. The contract for rebuilding was awarded to the Newport News Ship Building and Dry Dock Company, the total cost of which amounted to about $120,000. The result is that the vessel is now practically as good as new, and in many respects better than when she was originally designed. To reproduce her, not including her battery and equipment, would cost the Government at least $350,000, at the present prices for labor and material.

REBUILDING.

The old battery and the sponsons for the 4-inch guns were removed and the following guns are now mounted: Four 6-pounder Driggs-Schroeder, two 3-pounder Driggs-Schroeder, six 3-pounder Hotchkiss, two 1-pounder Hotchkiss, and two 1-pounder field guns.

The rig, which in September, 1896, had been changed from that of a barkentine to two pole mast, was replaced by that of an hermaphrodite brig. The dimensions follow:

Foremast:	
Lower mast, heel to hounds, feet and inches	44–0
Head, feet and inches	7–6
Fore topmast, feet and inches	38–9
Rake, degrees	5
Mainmast:	
Heel to hounds, feet and inches	46–0
Head, feet and inches	7–6
Main topmast, feet and inches	38–0
Rake, degrees	5½
Fore topgallant yard, length over all, feet and inches	27–0
topsail yard, length over all, feet and inches	40–0
yard, length over all, feet and inches	53–0

The standing rigging is all of galvanized iron.

All old wooden decks were removed and replaced with new. A new pilot house and chart room were fitted.

The protective deck over the engine and boilers was removed and trunks carried up to the spar deck, large openings being left for light and ventilation.

The bunker bulkheads in the engine room were removed and this space utilized for auxiliary machinery. A cross bunker was built between the engine and boiler space to make up the coal capacity lost by removal of the engine-room bunkers. At each side and directly outboard of this cross bunker fresh-water tanks are built in, with a total capacity of 4,200 gallons. The capacity of the lower bunkers is now about 117 tons, a little in excess of what it formerly was.

This arrangement has increased the floor space around the machinery. The fireroom is light and airy, and is 10 feet long by 19 feet wide. The new arrangement of the vessel is appended. Plates II and III.

BOILERS.

The two original boilers, of the horizontal, fire-tube gunboat type, had two corrugated furnaces each, with a separate combustion chamber for each furnace. The following are the principal data:

Number of boilers	2
Steam pressure, pounds	160
Length, feet and inches	17–0
Diameter, outside, feet and inches	8–9½
Grate surface, total for two boilers, square feet	87.75
Heating surface, total for two boilers, square feet	2,686.0
Heating surface ÷ grate surface	30.61
Furnaces, mean internal diameter, inches	41
Number of tubes, one boiler	314
Length of tubes between tube sheets, feet and inches	7–0
Outside diameter of tubes, inches	2
Area through tubes, one boiler, square feet	5.44
Grate surface ÷ tube area, square feet	8.06
Diameter of funnel, feet and inches	4–7
Height of funnel above grate, feet	40

These boilers were removed, together with their uptakes, funnel and all other fittings. They were replaced by Babcock & Wilcox boilers of the following dimensions:

Number of boilers	2
Drums, length of shells, feet and inches	7–4⅝
inside diameter, feet and inches	3–0
Drum plates, thickness, inch	0½
Heating surface, two boilers, square feet	3,825
Grate surface, two boilers, square feet	85
H. S. ÷ G. S.	45
Funnel, height above grates, feet and inches	48–0
inside diameter, feet and inches	5–0
Generating tubes, outside diameter, inches	2 and 4
thickness of 2-inch	No. 8 B.W.G.
4-inch	No. 6 B.W.G.
exposed length, feet	9
inclination of, degrees	15
number of 2-inch, one boiler	314
4-inch, one boiler	12
Height of boiler, base to drum center, feet and inches	9–5¼
casing top, feet and inches	10–1½
Width of boiler, feet and inches	7–3¼
Length of boiler at base, feet and inches	10–3¼

The pressure parts of these boilers are of mild steel and the tubes cold drawn. All details are up to the usual Government standard. The boilers are placed abreast, and well forward in the old boiler compartment. The gases discharge

into a single funnel. There is a space, 30 inches wide and of ample height, between the boilers, for passing coal from the forward bunkers.

FORCED DRAFT.

The original boilers were fitted with forced draft on the closed ash-pit system. Air was supplied by two fans bolted to the coal-bunker bulkheads and taking air from ducts over the tops of the boilers.

Diameter of cylinders (two), inches	$3\frac{1}{2}$
Stroke, inches	$2\frac{1}{2}$
Diameter of fan, inches	36
Width of fan, inches	12
Diameter of inlet, inches	22

These fans were removed and one fan of the Sturtevant make and of the following dimensions installed:

Diameter of cylinders (two), inches	5
Stroke, inches	4
Diameter of fan, inches	60
Width of fan at periphery, inches	17
Diameter of inlet, inches	42

The fan engine is located in the upper engine room, the fan being outside the engine-room casing on the berth deck. It takes air from the crew space direct and discharges through a 20-inch by 48-inch duct, below the berth deck, to the fireroom. Air valves and ducts are also connected to the fan casing and living quarters for exhausting vitiated air.

PROPELLING ENGINES.

The cylinders were $13\frac{1}{2}$, 21 and 31 inches diameter and 20 inches stroke. In order to increase the economy and steaming radius of the vessel, it was decided to raise the steam pressure from 160 to 215 pounds and cut down the high and intermediate cylinder diameters.

New H.P. cylinders, 12 inches diameter, and new M.P. cylinders, 19 inches diameter, complete with new pistons and

valves, were made for each engine, and a very light cut was taken through each L.P. cylinder. The piston rods and valve stems were either renewed or resurfaced and new packing for pistons, piston rods and valves stems was furnished.

New crossheads, crosshead pins and crosshead bearing brasses were made, and the bar guides were taken down and planed on all four faces.

The crank and thrust shafts were removed and trued up on all bearing surfaces, and brasses and bearings were refilled with white metal, refitted and lined up. The stern-tube and bracket bushings were restaved with lignum vitae. All eccentric sheaves were trued up and the straps remetaled and bored out. The reversing cylinders and gear were overhauled and the oil cushioning cylinders and hand pumps removed.

New balanced throttle valves were fitted in place of the old gridiron valves. New oil and water services, gauges and revolution counters were also fitted.

The following are the principal data for the main engines as now installed:

Number of engines	2
Cylinders, number for each engine	3
diameter of H.P., inches	12
M.P., inches	19
L.P., inches	31
Stroke of pistons, inches	20
Ratio of the net piston areas, at present	1 : 2.5 : 6.8
old engines	1 : 2.4 : 5.3
Valves, diameter of the H.P., inches	6
M.P. (two), inches	6
length of the L.P., feet and inches	2–0⅞
width of the L.P., feet and inches	2–3
Piston rods, diameter, inches	2½
Main steam pipe, diameter, inches	4½
Exhaust to M.P., diameter, inches	5¾
L.P., diameter, inches	7¾
condenser, diameter, inches	9¾
Clearance in H.P. cylinder (volumetric), per cent	11.67
M.P. cylinder (volumetric), per cent	12.0
L.P. cylinder (volumetric), per cent	7.1
Connecting rods, length, center to center, inches	40
diameter at upper end, inches	2½
lower end, inches	4

Crosshead, surface of slipper, square inches	71.8
pins, diameter, inches	3½
Crank shaft, diameter, inches	6
axial hole, inches	3
journals, diameter, inches	6
length, inches	5½
length of each section, feet and inches	2-7½
Crank pins, diameter, inches	6
length, inches	6
axial hole, inches	3
Thrust shaft, diameter, inches	5¾
axial hole, inches	2⅞
collars, diameter, inches	9
number for each shaft	7
surface, total for one engine, square inches	263.55
Line shaft, diameter, inches	6
axial hole, inches	3
Propeller shaft, diameter, inches	6
axial hole, inches	3

PROPELLERS.

The propellers are of manganese-bronze with three elliptical blades, and turn outboard. The blades can be set for pitches between 6 feet 9 inches and 8 feet 3 inches. The boss is secured to the shaft by a taper key and a composition nut. The end of the shaft is covered by the usual cap.

PROPELLER DATA FOR THE TRIAL.

Number propellers	2
Diameter over tips of blades, feet and inches	7-0
of hub, feet and inches	1-10
Pitch as set for trial, feet and inches	7-9
Helicoidal area, square feet	15.0
Projected area, square feet	10.66
Disc area, square feet	35.85
Pitch ÷ diameter	1.1
Helicoidal area ÷ disc area	0.418
Projected area ÷ disc area	0.297
Immersion of shaft centers at mean draught on trial, feet and inches	7-8½

CONDENSERS.

The main-condenser shell is cylindrical, of composition and in one piece. The circulating water enters at the bottom and leaves at the top, the water flowing through the tubes.

The tubes and tube sheets were removed and the shell thoroughly cleaned. The old tube sheets were replaced and new tubes and ferrules fitted.

Diameter of the shell, feet and inches	4-0
Length over heads, feet and inches	7-8½
Tubes, outside diameter, inch	0⅝
length between tube sheets, feet and inches	6-3½
number of	1,772
Cooling surface, square feet	1,824

There was, besides, an auxiliary condenser, with an air and circulating pump, all of the Davidson type. As valves were fitted in the exhaust pipes from the main engines, this condenser was deemed to be unnecessary and was removed from the vessel.

AIR PUMP.

The air pump, of the Davidson twin-cylinder type, was thoroughly overhauled.

Diameter of the steam cylinder, inches	8
water cylinders (two), inches	14
Stroke, common, inches	12

CIRCULATING PUMP.

The circulating pump, of the usual centrifugal type, driven by a vertical steam engine, was overhauled and put in good order.

Diameter of steam cylinder (one), inches	6
Stroke, inches	5
Diameter of impeller, inches	22
Width of impeller at hub, inches	4½
tips of blades, inches	2
Diameter of inlet nozzle, inches	8
outlet nozzle, inches	7

FEED AND FILTER TANK.

The old feed and filter tank was removed from the vessel and a new tank, placed in a more accessible position, is now located on the port side.

PUMPING OUTFIT.

The following pumping outfit was removed from the vessel together with the old pipe system.

......	Kind.	Cylinders.		Stroke.
		Steam.	Water.	
		inches.	*inches.*	*inches.*
Main feed	vertical single	9	5½	10
Auxiliary feed	vertical single	9	5½	10
Water service	vertical single	9	5½	10
Fire and bilge	vertical single	12½	7¾	12
Distiller circulating	vertical single	5½	5	8
Evaporator feed and tank	vertical single	3	1¾ 1½ tank.	4

The new pumps fitted are as follows:

......	Kind.	Cylinders.		Stroke.
		Steam.	Water.	
		inches.	*inches.*	*inches.*
Main feed	horiz'l duplex	7½	4½	10
Auxiliary feed	horiz'l duplex	8	5	12
Water service	vertical single	7	7	10
Fire and bilge	vertical single	12	7	12
Distiller circulating	vertical single	4½	6	6
Evaporator feed	vertical single	3½	2⅛	4

The feed tank and horizontal pumps are placed well out in the wings in the space formerly occupied by the engine-room coal bunkers.

FEED-WATER HEATER.

A new feed-water heater with spiral tubes and about 90 square feet of heating surface was installed, the heating agent being the exhaust steam from the auxiliaries. No heater was fitted in the old feed system.

DISTILLING APPARATUS.

The old Baird evaporator, No. 3, Type C, being worn out and beyond repair, was removed from the vessel. A new evaporator and distiller of the Quiggins type, with spiral tubes, is now installed. The maximum capacity is 2,000 gallons per 24 hours.

OFFICIAL TRIALS UNDER FORCED DRAFT.

Draught, molded, at beginning, forward, feet and inches
aft, feet and inches
mean, feet and inches
end, forward, feet and inches
aft, feet and inches
mean, feet and inches
mean, for trial, feet and inches
Displacement at mean draught as above, tons
Average speed in knots per hour
Slip, mean of both screws, in per cent. of their own speed
Revolutions of the main engines, per minute
mean of both engines
Piston speed, feet per minute
Steam pressure at boilers (gauge)
M.P. receiver (absolute)
L.P. receiver (absolute)
Vacuum in condenser (inches of mercury)
Opening of throttle
Cut off, decimals of stroke from beginning (mean of both ends), H.P.
M.P
L.P.
Double strokes of air pump, per minute
feed pump, per minute
Revolutions of circulating pump, per minute
Temperature, engine room, degrees Fahrenheit
injection, degrees Fahrenheit
discharge, degrees Fahrenheit
hot well, degrees Fahrenheit
feed, degrees Fahrenheit
fireroom, degrees Fahrenheit
Revolutions of blowers, per minute
Mean effective pressure, H.P. cylinder, pounds
M.P. cylinder, pounds
L.P. cylinder, pounds
referred to L.P. cylinder, pounds
Indicated horsepower: H.P
M.P
L.P
Collective, each main engine
both main engines
Air pump
Circulating pump
Feed pump
Blowers (estimated)
Other auxiliaries (estimated)
Collective, all auxiliaries
all machinery in operation
Indicated thrust, main engines only, pounds
per sq. ft. developed area of propeller (15 sq. ft.), pounds
inch of thrust bearing (263.55 sq. inches), pounds
Square feet of cooling surface in condenser per I.H.P
heating surface in boilers per I.H.P
I.H.P. per square foot of grate surface
Kind of coal

U. S. S. "BANCROFT." Official trial, Jan. 26, 1893.			U. S. S. "ITASCA." Official trial, June 14, 1907.		
Starboard.		*Port.*	*Starboard.*		*Port.*
......	11-00	......	......	9-05	......
......	12-00	......	......	12-06	......
......	11-06	......	......	10-11½	......
......	10-10	......	......	9-04	......
......	11-11	......	......	12-05	......
......	11-04½	......	......	10-10½	......
......	11-05½	......	......	10-11	......
......	832.0	......	......	775.0	......
......	**14.87**	......	......	**15.28**	......
......	15.5	......	......	17.14	......
223 09	......	221.77	242.25	......	240.0
......	222.39	......	......	241.125	......
743.5	......	739.2	808.5	......	800.0
......	167.6	......	......	206.13	......
64.32	......	65.76	99.7	......	96.7
24.1	......	25.76	25.7	......	26.7
......	26.34	......	......	23.33	......
Wide.	......	Wide.	Wide.	......	Wide.
.72	......	.72	.68	......	.68
.72	......	.72	.70	......	.70
.72	......	.72	.70	......	.70
......	75.5	......	......	50.0	......
......	35.25	.....	.....	31.0	......
......	236.31	......	......	354.5	......
......	76.2	......	......	90.0	......
......	44.0	......	......	66.0	......
......	87.7	......	......	......	......
......	90.6	......	......	112.0	......
......	119.9	......	......	226.0	......
......	77.5	......	......	93.5	......
574.0	......	580.0	......	513.0	......
61.98	......	65.87	91.0	......	103.25
21.77	......	22.93	42.9	......	39.3
13.98	......	12.48	13.3	......	14.25
34.61	......	35.28	42.7	......	44.1
197.0	......	208.04	246.6	......	276.8
169.09	......	176.52	294.9	......	267.7
222.33	......	210.36	244.7	......	259.7
588.42	.....	594.92	786.2	.....	804.2
......	1,183.34	......	......	1,590.4	......
......	5.09	......	......	3.4	......
......	1.35	......	......	3.0	......
......	1.7	......	......	2.2	......
......	12.0	......	......	15.0	......
......	9.5	......	......	10.0	......
......	29.64	......	......	33.6	......
......	**1,212.98**	......	......	**1,624.0**	......
11,232.0	......	11,422.0	13,819.0	......	14,268.0
748.7	......	761.4	921.26	......	951.2
42.5	......	43.22	52.43	......	54.14
......	1.54	......	......	1.123	......
......	2.18	......	......	2.355	......
......	13.82	......	......	19.2	......
Selected Pocahontas.			Georges Creek, mostly slack.		

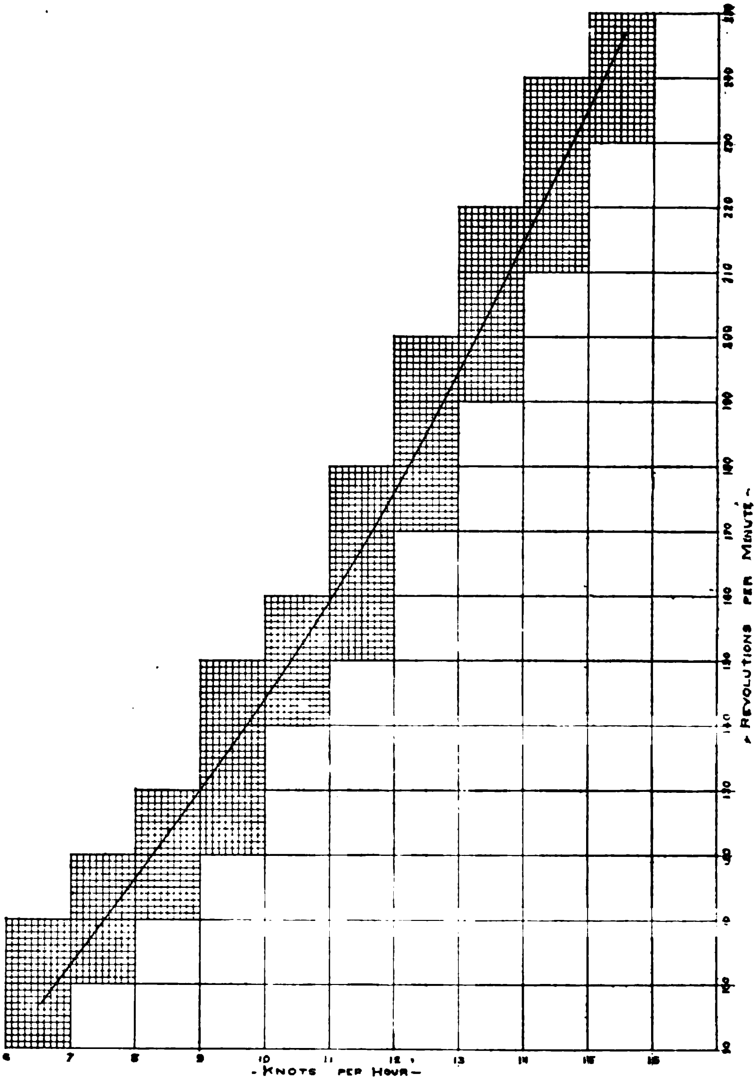
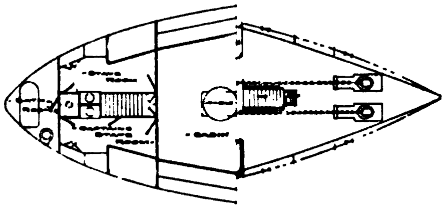
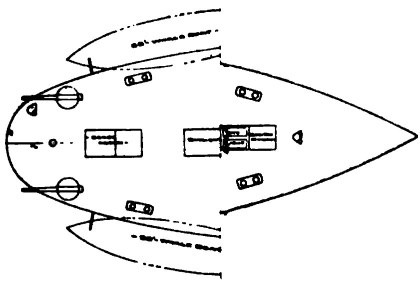


Plate I.—STANDARDIZATION CURVE, U. S. S. "ITASCA."





VENTILATING APPARATUS.

In the engine room there was formerly a small steam-driven fan for ventilating the living quarters. This fan was removed and the air ducts connected to the forced-draft fan on the berth deck.

OTHER AUXILIARIES.

The steering engine is placed well aft, near the tiller and beneath the protective deck. It is of the Williamson type, with steam cylinders 6 inches by 6 inches. This engine was overhauled. The windlass engine, of the Providence type, was fitted with new brasses and operating valves.

ELECTRIC PLANT.

The electric plant consisted of two 4-kw. Thompson-Houston dynamos delivering current at a pressure of 80 volts. They were driven by Armington & Sims' vertical engines at 500 revolutions per minute. Each engine had two cylinders, 5 inches diameter and 3 inches stroke. These units being worn out, were removed.

The new generating set is of the General Electric make. The generator has six poles and delivers 15 kw. at a pressure of 125 volts at the switchboard. The engine is driven by a single steam cylinder, 8 inches diameter and 6 inches stroke. The set averaged 408 revolutions per minute on trial.

The old wiring and fixtures were removed and entirely new systems installed. All wires are in metal conduits and most of the fixtures are steamtight. The vessel is fitted with 202 sixteen-candlepower lamps, a telephotos, truck lights and a General Electric 18-inch searchlight. New general alarms and call bells were fitted throughout.

THE TRIALS.

On June 13, 1907, the dock trial was made and, everything working successfully, the standardization and four-hour trials were run on the following day.

The vessel left the ship yard at 10·40 A. M. and made several runs over the measured mile off Newport News. The

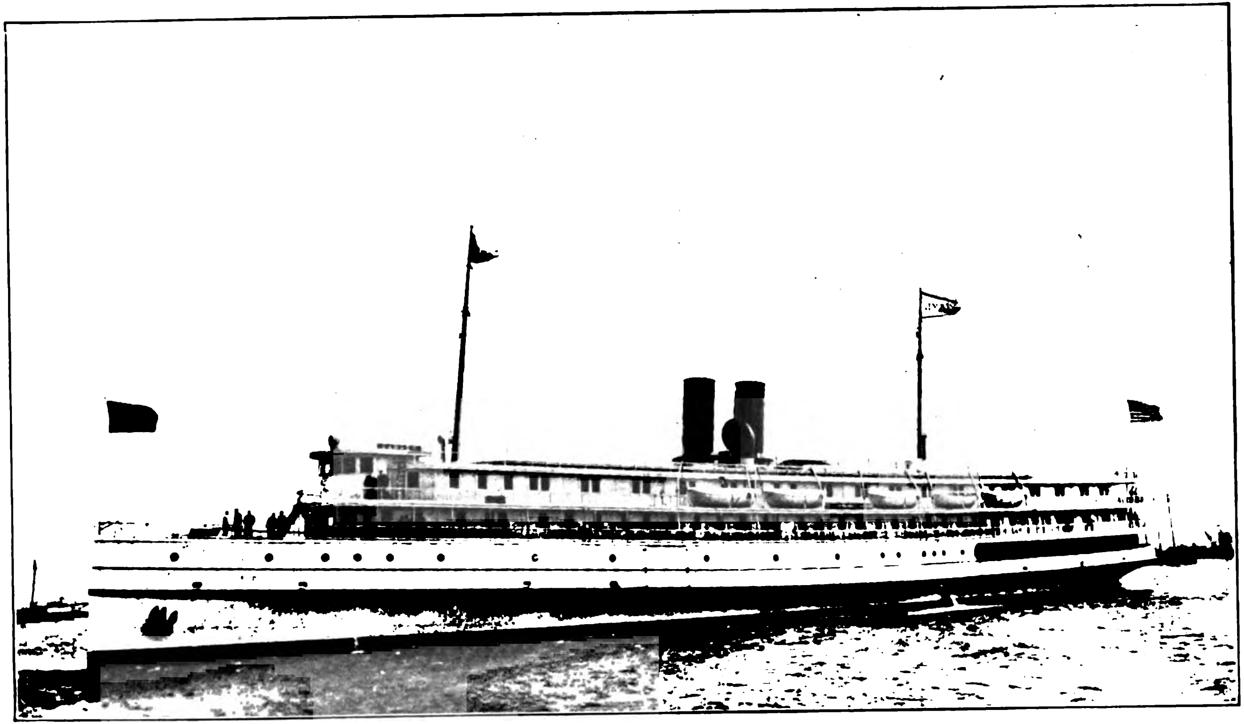
results checked exactly with the original standardization curves.

The endurance trial, under forced draft, commenced at 1·00 P. M. and ended at 5·00 P. M. There was a moderate beam sea and a fresh breeze. The course was from the Thimble Light off Old Point Comfort to the sea buoy off Cape Henry. The vessel was trimmed too much by the stern, and though the displacement was slightly less than on the original trial, the indicated horsepower for any given speed of the propelling engines was from five to ten per cent. higher than on the original trial. The trial was successful in every way. The boilers steamed freely in spite of the fact that the coal was of very poor quality. The water level in the boilers was steady at all times.

The main engines and auxiliaries performed in a satisfactory manner. No water was used on any of the bearings. The dynamo was in operation during the entire run.

The turns were made at full speed at each end of the course, both throttles being wide open for the entire four hours. Live steam was frequently admitted to the M.P. and L.P. receivers. On this account it is impossible to make a comparison, for steam consumption, between the vessel's performance with the old and new cylinder ratios.

The *Itasca* is now on a European cruise, and it is expected that sufficient data will have been obtained, by the time she returns, to make an interesting comparison. A synopsis of the vessel's official trials of January 26, 1893, and June 14, 1907, is appended, Table and Plate I.



S. S. "CAMDEN."

THE TURBINE STEAMSHIP *CAMDEN*.

DESCRIPTION AND TRIALS.

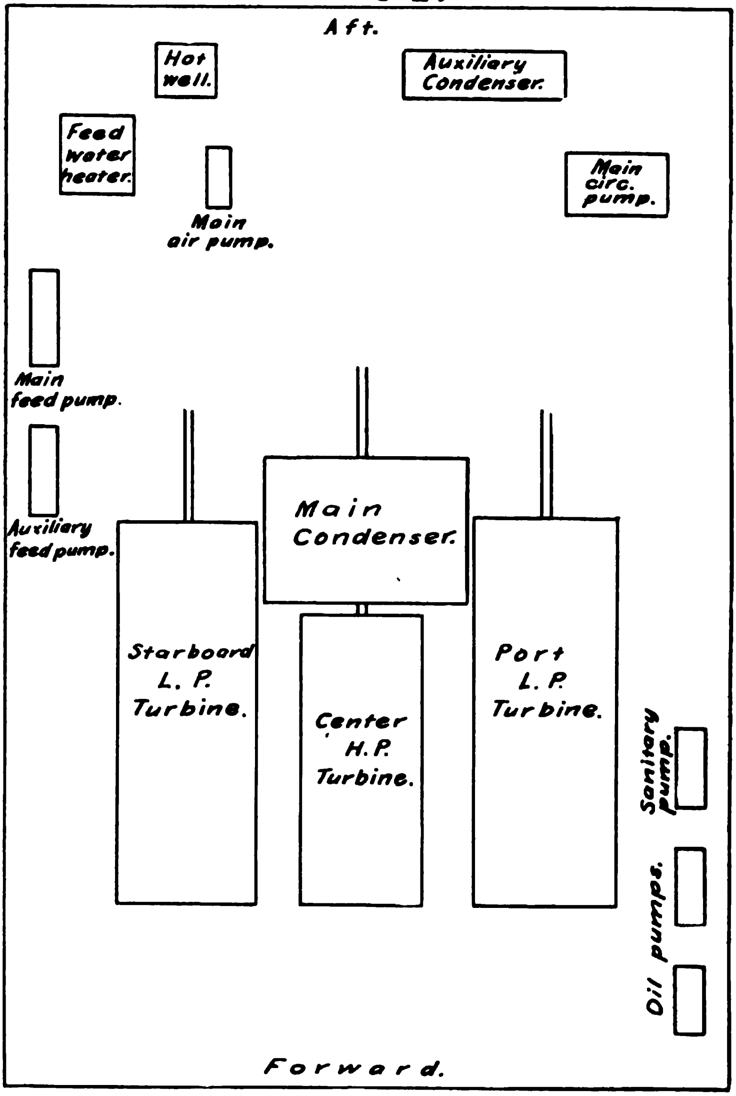
BY LIEUT. A. F. H. YATES, U. S. N., MEMBER.

The *Camden*, built by the Bath Iron Works, Bath, Me., is one of two passenger ships contracted for by the Eastern Steamship Company for service between Boston, Mass., and Bangor, Me. The contract for the construction of the *Camden* was concluded June 1, 1906, and provided for the delivery of the vessel on or before June 15, 1907, a bonus of $300 per day being allowed for delivery in advance of the date specified. The guaranteed speed of the vessel was an average of at least seventeen knots an hour on six double runs of the vessel over her passenger route. Owing to market conditions in the United States the fulfillment of the contract within the time specified would have been extremely doubtful had not contracts for the machinery, steel castings and steel drums been promptly placed in England. The principal steel castings and drums were delivered within five months from the date of the contract.

The keel of the *Camden* was laid on July 19, 1906, the hull launched on February 14, 1907, the trials of the machinery finished on May 23, 1907, and the vessel delivered on May 31, 1907.

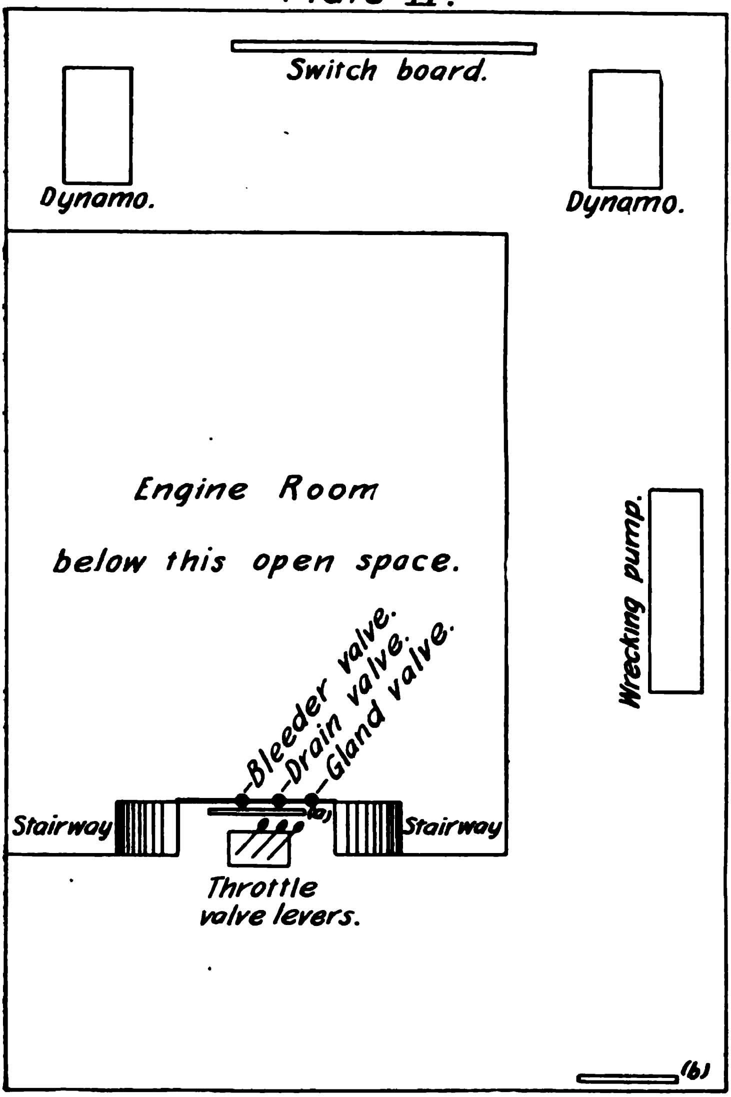
The machinery trials were held at Boothbay Harbor, Me., and were of particular interest, the *Camden* being one of the first vessels to be fitted in the United States with a Parsons turbine installation. The installation on the *Camden* is made up of one center and two wing turbines of about 4,000 brake horsepower, making upwards of five hundred revolutions per minute. The center turbine is the high-pressure ahead turbine, and takes its steam direct from the boilers. The wing turbines are low-pressure and are run ahead on the

Plate I.



SHOWING ARRANGEMENT OF MACHINERY IN ENGINE ROOM, STEAMSHIP "CAMDEN."

Plate II.



(*a*) Frame for the following gauges: main steam pressure; starboard astern pressure; port astern pressure; starboard ahead pressure; port ahead pressure; starboard revolution indicator; center revolution indicator; port revolution indicator; vacuum, main condenser; clock.
(*b*) Frame for the following gauges: auxiliary steam pressure; auxiliary exhaust pressure; steam heat; oil pressure; gland pressure.

SHOWING ARRANGEMENT OF ENGINE-ROOM PLATFORM OVERHEAD, STEAMSHIP "CAMDEN."

exhaust steam from the center (high-pressure) turbine. No live steam is used in running the wing turbine ahead, the only steam admittance being through the thirteen-inch receiver pipes from the center turbine. The wing turbines are fitted with separate astern turbines, which, when in use, are run on live steam from the boilers. Although there is an auxiliary condenser installed it is only used in part, the exhaust from the wing turbines and from all auxiliaries, when under way, being connected with the main condenser, which has a Parsons augmentor attached.

An interesting feature of the installation is the method of controlling the pressure in the glands. Auxiliary exhaust steam is used entirely, and the pressure on the various glands (from 2 to 3 pounds) remains the same, by reason of an equalizing pipe which connects all of the glands. The amount of steam admitted is regulated by watching the vapor which leaks from the glands into the atmosphere, the regulating valve being so located that the man operating it can see all of the glands.

Lubricating oil is pumped to the main bearings through a cooler, and drains back to the main oil tank.

The following is a description of the vessel and machinery. Plates No. 1 and No. 2 show the arrangement of machinery in the engine room and on the overhead platform, respectively.

HULL DATA.

Load draught, forward, feet and inches	9-0⅛
aft, feet and inches	9-5½
Displacement on load draught, tons	1,824
on trial draught, tons	1,824
per inch at L.W.L., tons	22
Length over all, feet and inches	335-4
on L.W.L., feet and inches	320-0
Beam over guards, feet and inches	54-2
on L.W.L., feet and inches	40-0¾
molded, feet and inches	40-0
Area L.W.L. plane, square feet	9,200
Capacity of coal bunkers, tons	200
engine-room feed tanks, tons	4
reserve feed water compartments, tons	32

MAIN ENGINE DATA.

Diameter of starboard L.P. rotor, inches	58
center H.P. rotor, inches	41
port L.P. rotor, inches	58
astern rotor, inches	47
No. of expansions, starboard rotor	8
center rotor	4
port rotor	8
astern rotor	4
Total intended brake horsepower	4,000
Expected mean speed, knots	17

SHAFTING AND BEARINGS.

Line shaft, center, steel, 106 feet long, solid, diameter, inches	6
wing, steel, 64 feet long, solid, diameter, inches	6
Propeller shaft, center, steel, 16 feet long, solid, diameter, inches	6¼
wing, steel, 36 feet long, solid, diameter, inches	6¼
Collars on shaft:	
Number	16
Thickness, inch	0¼
Space between, inch	0⅞
Outside diameter, inches	10
Inside diameter, inches	6¼
Thrust shoes, number	16
effective thrust surface, square inches...top, 287; bottom,	290
Line shaft bearings: 4 starboard, 4 port, 5 center; length, each, ins.	9
Stern-tube bearings, forward and after, length, each, inches	36
After strut bearing, length, inches	36

CONDENSER AND FEED HEATER DATA.

One main surface condenser, surface, square feet	4,481
auxiliary condenser, cooling surface, square feet	140
augmenter condenser, cooling surface, square feet	135
pressure condenser (auxiliary exhaust), cooling surface, sq. ft.	321
oil cooler, condenser type, cooling surface, square feet	132

PROPELLER DATA.

Three solid Hyde manganese-bronze:	
Number of blades	3
Diameter, feet and inches	5-00
Pitch, feet and inches	4-06
Area 3 blades, square feet	8.8
helicoidal, square feet	10
Height of lower tip of blade above keel, inches	18
Immersion of upper tip of blade, inches	33

BOILER DATA.

Four three-furnace, Scotch boilers, 45-inch Morison furnaces:	
Designed working pressure, pounds	150.0
Pressure test, pounds	225.0
Ratio, G.S. to H.S.	1 to 44.3
Grates, length, feet and inches	5-8
width (total width one boiler), feet and inches	11-3
Diameter of boilers, feet and inches	14-6
Length of boilers, feet and inches	12-0¾
Grate surface one boiler, square feet	64.0
four boilers, square feet	256.0
Heating surface, one boiler, square feet	2,835.0
four boilers, square feet	11,340.0
Boiler tubes of lapwelded charcoal iron, 2¼ inches outside diameter, length, feet and inches	8-7¼
Ellis and Eaves' induced draft:	
Height of smoke pipes above grates, feet	60
Area of smoke pipe, square feet	28.0

DATA OF PUMPS AND AUXILIARY ENGINES.

		Steam	×	Water	×	Stroke.
1 Main air	Blake vert. turn beam, independent	12	×	25	×	18
1 Main circ.	Cent., two 12-in. suction, one 16-in. disch., Bath Iron Wks. Engine,	8		...		8
2 Main feed	Blake horizontal duplex	8	×	6	×	12
4 Boiler circ.	Blake horizontal duplex	4½	×	2¾	×	4
2 Oil pumps	Blake horizontal duplex	6	×	5¼	×	6
1 Donkey F. & B.	Blake horizontal duplex	10	×	8½	×	12
1 Sanitary	Blake horizontal duplex	6	×	5¼	×	7
1 Aux. cond.	Blake combined air and circ.	5½	×	6	×	7
1 Donkey feed.	Blake horizontal duplex	4½	×	2¾	×	4
1 Fresh water	Blake horizontal simplex	3½	×	3½	×	4

2 Dynamo engines, 35 kw., General Electric, turbo generators, Curtis.

2 Forced-draft blowers, 84-inch diameter fans, 17-inch face, 5 inch × 5 inch, Bath Iron Works Engine.

On the morning of May 23, 1907, at 9 o'clock, the *Camden* left the dock at the Bath Iron Works, Bath, Me., and proceeded to the trial grounds at Boothbay Harbor, Me. Four standardization runs were held over the measured mile, under natural draft, followed by two runs under forced draft. At the conclusion of these runs a two-hour run was held under forced draft. The vessel then returned to Bath, Me., and secured at the dock at 3·30 P. M. During the first hour of this latter run a grinding noise was heard for several seconds.

STANDARDIZATION TRIALS OF S. S. *CAMDEN* ON COURSE OFF SOUTHPORT, ME., ON MAY 23, 1907.

Number and direction of run.	Elapsed time.	Steam pressure.				Vacuum, in inches.	Auxiliary exhaust, pounds.	Revolutions.				Torsion meter readings.			Temperature, degrees Fahr.				Speed, in knots.	Shaft horsepowers.			
		At engine ab. atmosphere.	Center turbine.	Starboard turbine.	Port turbine.			Starboard shaft.	Center shaft.	Port shaft.	Mean 3 shafts.	S	C	P	Injection.	Discharge.	Hot well.	Feed.		Starboard shaft.	Center shaft.	Port shaft.	Total.
1 N	4.556	142	35	*3.0	*4.0	28.2	1.5	357.7	346.8	357.7	354.1	.28	.34	.33	43	60	68	193	13.17	367	436	432	1235
2 S	4.193	143	55	*3.0	†1.0	28.2	1.5	410.2	395.9	407.8	404.6	.36	.41	.38	44	66	68	190	14.31	544	600	568	1712
3 N	3.646	137	79	†7.6	7.3	27.9	1.0	466.3	452.5	463.5	460.8	.48	.50	.48	42	68	68	190	16.46	811	837	815	2463
4 S	3.533	134	100	11.0	12.0	27.6	2.5	495.3	481.2	498.2	491.6	.58	.60	.58	42	74	74	178	16.98	1053	1068	1059	3180
5 N	3.123	152	139	17.7	20.0	27.7	3.0	563.5	544.3	560.4	556.1	.72	.76	.74	42	74	74	174	19.21	1487	1532	1521	4540
6 S	3.190	152	139	18.0	20.0	28.0	3.0	558.0	542.3	561.1	553.8	.71	.78	.73	42	74	74	170	18.81	1447	1566	1502	4515

* Inches vacuum. † Pounds.

Run.	Counter Readings.								
	Starboard.			Center.			Port.		
	On.	Off.	Diff.	On.	Off.	Diff.	On.	Off.	Diff.
1 N	25,000	26,630	1,630	32,230	33,810	1,580	51,610	53,240	1,630
2 S	30,670	32,390	1,720	37,750	39,410	1,660	57,210	58,920	1,710
3 N	36,470	38,170	1,700	43,470	45,120	1,650	63,030	64,720	1,690
4 S	42,730	44,480	1,750	49,540	51,240	1,700	69,240	71,000	1,760
5 N	48,550	50,310	1,760	55,260	56,960	1,700	75,130	76,880	1,750
6 S	54,950	56,730	1,780	61,430	63,160	1,730	81,500	83,290	1,790

Run.	Auxiliaries, Double strokes.				Revs. circ. pump.
	Air.	Feed	Oil.	Sanitary	
1 N	50	22	19	24	192
2 S	52	22	20	24	192
3 N	52	24	19	24	188
4 S	48	26	20	24	186
5 N	52	34	20	24	192
6 S	52	32	19	24	186

$$H.P. = \frac{T \times R \times 19{,}455}{5{,}255}$$
for center shaft.

$$H.P. = \frac{T \times R \times 19{,}270}{5{,}255}$$
for wing shafts.

T = torsion meter reading.

R = revs. per min.

19,455 and 19,270 are calibration constants.

U. S. S. *CAMDEN*, COURSE OFF SOUTHPORT, ME., ON MAY 23, 1907.

TAKEN FROM CURVES.

Speed, knots.	Revolutions per minute—mean of three shafts.	Steam pressure, H.P. turbine.	Total shaft H. P.—three shafts.
5	128	8	200
9	240	18	575
13	357	40	1,275
15	419	61	1,850
17	485	95	2,855
19	555	144	4,575

DATA TAKEN DURING LAST HOUR OF TWO-HOUR SEA RUN.

Time.	Auxiliaries, dbl. strokes.					Steam at turbine.				Temperatures.					Counter readings.			Revs. per minute.			Torsion meter readings.			Shaft horse-powers.			
H. M.	Air pump.	Feed pump.	Oil.	Circulating.	Boiler steam.	Starboard.	Center.	Port.	Vacuum.	Injection.	Discharge.	Hot well.	Feed.	Aux. exhaust, lbs.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Starboard.	Center.	Port.	Totals.
1 00	47	34	20	190	149	16.5	140	19.5	27.6	43	75	75	180	1.5	632,410	609,930	606,260	...	...	...	...	...	...	...	...	...	...
1 10	46	29	20	190	148	16.5	137	19.3	27.9	43	75	73	176	2.0	638,210	615,500	612,050	580	557	579	.70	.76	.70	1,488	1,567	1,485	4,540
1 20	45	31	20	188	142	15.5	132	18.1	27.8	43	74	72	176	2.0	643,720	620,790	617,510	551	529	546	.67	.74	.67	1,353	1,449	1,341	4,143
1 30	45	30	14	180	145	15.9	134	18.8	27.9	43	76	73	178	3.5	649,280	626,050	623,040	556	526	553	.68	.76	.68	1,386	1,480	1,379	4,245
1 40	44	28	20	174	142	15.4	132	18.2	27.9	43	78	74	178	3.0	654,870	631,410	628,610	559	536	557	.71	.76	.68	1,455	1,508	1,388	4,351
1 50	44	28	21	168	137	14.0	126	16.5	27.6	43	75	73	180	3.0	660,280	636,600	634,010	541	519	540	.66	.72	.64	1,309	1,383	1,267	3,959
2 00	37	28	20	168	137	15.0	126	17.1	27.3	43	75	73	180	1.5	665,750	641,800	639,360	547	520	535	.66	.72	.62	1,323	1,386	1,216	3,925
Avg...	44	29	20	180	143	15.5	132	18.2	27.7	43	75	73	178	2.4	...	...	...	556	531	552	.68	.743	.665	1,385	1,463	1,346	4,194*

*Speed corresponding, 18.6 knots.

This was attributed to the astern dummy in the port low-pressure turbine. The turbines were slowed down and worked up again gradually and the noise ceased. The turbines continued to run smoothly during the remainder of the run, and it was concluded that no injury had resulted. This conclusion was borne out several days later when the turbine casing was lifted and a thorough examination made. Aside from this occurrence, the machinery worked most satisfactorily throughout all of the runs. No vibration was noticed in the shaft alley.

The two-hour run was primarily for determining the coal consumption, but, owing to the fact that the turbines had to be slowed down, the data secured were unreliable and are not noted herein. The brake horsepower was determined through the use of a Denny & Johnson torsion meter.

The data were secured during the vessel's trials.

Plate No. 3 shows a revolution and horsepower curve prepared from the above data.

At the conclusion of the trial the telegraph was thrown from full ahead to full astern. The time consumed in operating the turbines was inappreciable. The vessel came to a dead stop in two minutes and thirty-four seconds.

MODERN ARMOR AND ARMOR-PIERCING PROJECTILES.

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The problem of armor and armor-piercing projectiles is one of the most interesting and complex in the study of ballistics. Its peculiar difficulty lies in the fact that information concerning the curious phenomena that have from time to time been recorded has been information of a more or less qualitative kind, obtained under conditions of great dissimilarity, when the observation of the particular facts recorded was not the object for which the experiments were primarily devised. Organized trials of standard plates under standard conditions would involve such expense that private firms can hardly be expected to undertake them, and, as a consequence, most of our information concerning the behavior of armor under attack has been obtained from proof trials for acceptance or premium, and, naturally, under these circumstances the commercial aspect has predominated.

It thus comes about that our knowledge of this important branch of ballistics is confined to general comprehensive statements which simply indicate the tendency of a series of isolated observations. So long as the materials and processes are more or less secret and confidential, and the value of a plate is determined by its ability to cope with a special class of projectiles, so long must any solution of the problem of armor and projectiles be merely tentative and unsatisfactory. Nevertheless, there are certain aspects of the subject of which it may be useful to define the general scope and character, although from the gunner's point of view they may not appear to be intrinsically the most significant.

ARMOR PLATES.

Modern armor is universally made of one or other of the ferro-alloys, usually nickel-chrome, low in carbon, with a small percentage of manganese, tungsten, &c.; the proportions are usually considered a trade secret, and in all probability vary with the thickness of the plate. A good average Krupp plate contains $3\frac{1}{2}$ per cent. nickel, $1\frac{1}{2}$ per cent. chromium, $\frac{3}{10}$ per cent. carbon, and $\frac{7}{10}$ per cent. manganese. The plates may be cast, up to 6 inches or 7 inches in thickness, or forged; and they may be cemented or non-cemented. Cementation results from a superficial carburization, which enables a glass-hard face to be obtained without in any way affecting the tough qualities of the remainder of the plate. This effect is produced by subjecting the face to a temperature suitable for the solution of carbon placed in contact with it, the subsequent tempering permitting the production of a surface of any degree of hardness. The depth to which the cementation has effect depends upon the composition of the plate, its temperature, the time of exposure and the nature of the carburizing process. The solid carbon process of Harvey, which gave a face composition of as much as $\frac{7}{10}$ to $\frac{8}{10}$ per cent. of carbon, consisted in covering the furnace base by a few inches of sand before the plate was laid on it. The upper face of the plate was then covered with a mixture of animal and vegetable charcoal about a foot in thickness, over which a covering of tiles was put, the doors of the furnace being then bricked up and a high temperature maintained for many days.

The modern method of Krupp consists in keeping ordinary illuminating gas or petroleum vapor under pressure in contact with the heated plate, a process that was covered by Thwaite's English patent in 1893, and said to possess the advantages of more uniform and definite action, combined with lower temperature, shorter time and reduced cost.[1] In general the cementing materials belong to three classes—those acting by the formation of carbonic oxide, those acting by means of a

[1]Castner. "Stahl und Eisen," September, 1895.
Edwards. "Armor Plate." British Association, August, 1906.

cyanide and those employing hydrocarbons. Experiments made with pure carbon in a vacuum have demonstrated that no cementation takes place unless air is present and carbonic oxide formed. In the case of the cyanide there is simple decomposition, in which the carbon is liberated to be dissolved in the steel; while, when hydrocarbons are employed in the form of illuminating gas, petroleum vapor, &c., the carbon is released by dissociation.[2] The chemical and structural changes effected by cementation have been studied by Arnold;[3] and Charpy has published a most complete account of some quantitative researches on the cementation of steel with different carburizing materials in electric furnaces at definite temperatures.[4]

Cementation seems to be at its best for about a 6-inch plate, the skin effect, after the differential heat treatment, extending to about 2 inches, and the 2 to 1 proportion between tough back and hard face represents good practice. It has been found necessary that the steel for cemented and water-hardened face armor must be much more carefully selected and treated than was that for the early chilled-steel armor; and it is eminently necessary that the plate should gradually grade from the face quality to the back quality, otherwise under attack the hard skin flakes off or cracks in much the same way as did the old compound plates. Most makers object to the cementation of thin plates, the general experience being that, unless the plate is small and flat, cementation is not a success, warping and cracking of the plate being unavoidable. Krupp, however, professes to have solved the problem of manufacturing thin curved plates possessing all the features of Kruppized armor, as they are used, for instance, in the turret bases of cruisers.[5] Below 4-inch or 5-inch, which is usually considered to be the limit of cemented armor, the

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[2] Guillet. "La Cementation des Aciers au Carbone et des Aciers Speciaux." Societé des Ingenieurs Civils de France, February, 1904.

"Nouvelles Recherches sur la Cementation des Aciers au Carbone et des Aciers Speciaux." "Comptes Rendus," June, 1904.

[3] Arnold. "Micro-chemistry of Cementation." "Proceedings," Iron and Steel Institute, August, 1898

"Diffusion of Elements in Iron" "Proceedings," Iron and Steel Institute, May, 1899.

[4] Charpy. "Sur la Cementation du Fer." "Comptes Rendus," April, 1903.

[5] Krupp. "The Engineer," January 31st, 1902.

Charpy high-nickel cemented plates appear to be the best, as they can be made down to about 1 inch in thickness, and are thus of great value for destroyers and the shields of light Q.F. guns.[6] Usually the 5-inch limit of cemented plates marks the limit for non-cemented plates, the treatment of which is identical with that of the ordinary cemented plate, excepting that there is no superficial treatment, and the plate retains a tough quality throughout. We appear to be the only people to make non-cemented armor for ordinary purposes. Within recent years both Krupp and Hadfield have successfully produced cast-steel armor up to 6-inch and 7-inch in thickness, with a factor of merit approaching 2; this nature of armor seems particularly suitable for ammunition tubes, gun shields, &c.

The manufacture of a modern cemented plate involves a large number of costly processes, and a series of special heat treatments.[7] The steel ingots weighing from 60 to 100 tons are made in open-hearth furnaces, with either acid or basic linings, and chemical analyses are made to ensure the composition of the steel being within the limits laid down by practice. The most important point in the subsequent treatment of the ingot is the limited range of temperature at which it is possible to get satisfactory results, for the nickel-chrome alloy as forged may, by suitable heat treatment, have its tenacity increased from 40 to 90 per cent., and its ductility from 8 to 30 per cent., &c.[8] For armor plate reheating the temperature must be known to within 10 degrees or 20 degrees centigrade, and one of the simplest foundry methods of temperature measurement by an iron ball and water calorimeter appears to afford sufficiently accurate indications for practical purposes.[9] For a comparison of the value of various pyrometric instruments for industrial purposes reference may be made to the works of Le Chatelier, to a discussion at the Iron and Steel Institute,[10] and to a paper by Whipple.[11]

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[6] Tresidder. "Brassey's Annual," 1905. Pp. 366–371.

[7] Johns. "The Production and Thermal Treatment of Steel in Large Masses." "Proceedings," Iron and Steel Institute, 1903.

[8] Hadfield. Presidential Address "Proceedings," Iron and Steel Institute, 1905.

[9] Gledhill. "Proceedings," Iron and Steel Institute, May, 1904.

[10] Exhibition of Pyrometers. "Proceedings," Iron and Steel Institute, May, 1904.

[11] Whipple. "Pyrometrics for Industrial Purposes." Cleveland Institution of Engineers, December 5th, 1904.

The steel ingot is left in the molds, but is not allowed to get cold, it being removed to a forge, and heated by gas for many hours. It is then forged, sometimes at the rolling mills, sometimes under a press, and sometimes partly at the rolls and partly at the press. The correct temperature for working the metal at the rolling mills is secured by the use of pyrometers; micro tests and cooling curves are made, and the critical points determined in the laboratory. When roughly forged the ingot goes back to the reheating furnace for several hours, after which it is rolled roughly to dimensions. The plate is then supported on knife edges, some feet above the ground, so as to admit of a free supply of air, and when the plate has cooled to a certain temperature it is put into a special low-temperature furnace, after which it is softened by immersion in water. Pneumatic scaling hammers then remove the scale and the plate is machined, the surface being carefully examined for cracks. The plate is then taken to the cementing furnace, and when in pairs the carburizing medium is sandwiched between them. The time during which the plate is undergoing cementation varies from two to three weeks, and pyrometric records are either taken at frequent intervals of about 15 minutes, or an automatic arrangement is used which rings a bell when the temperature falls too low. When the process is complete the gas is shut off and the furnace allowed to cool for another week, after which the plate is reheated and plunged into rape or other oil to harden, the time of the last reheat being determined by the size, weight and thickness of the plate. The plate is again heated and plunged into a circulating cooling solution, after which it goes to the presses and is bent to the molds, the latter having been taken directly from the ship, or, in the latest practice, from the ship's drawings.[12] The plate is again heated, immersed, and allowed to cool. It is then sent to the machine shops and finished to size. When the final differential heat treatment of the cemented plate has taken place, the face is so hard that, without special local treatment, holes

[12] White. "Modern Warship." Cantor Lectures. Society of Arts, 1906.

cannot be drilled or tapped. To overcome this difficulty, it has been usual either to protect the surface of the plate during cementation in places where holes were likely to be required, or else to make accurate drawings and patterns, and to do all work on the plate before the final hardening. The first method is attended by the difficulties arising from errors in the foundry and the drawing office, and also that no certain means of protecting patches of the plate has been discovered. The second method, although slow and costly, is common in this country, the plate being bored, tapped and countersunk, and the holes filled with clay, before the plate is hardened. Attempts have also been made locally to anneal the plate by means of the blow pipe or electric arc, but without conspicuous success. In America, and to some extent in this country and Germany, electric annealing currents are passed through the plate by pressing two copper contacts, cooled by circulating water, against the plate, and this method has also had successful application to the softening of strips of metal for the removal of slots for sights.[13]

The plate, having been finally prepared, now undergoes what is known as the differential heat treatment, in which the face is raised to a higher temperature than the back, so that, whilst on sudden cooling the face is rendered glass-hard, the back retains its essential characteristics of toughness. The plate then goes back to the machine shops, and the surface curves are tested to see whether they correspond with the molds. If the plate has warped it is reheated, and the differential heat treatment repeated after the requisite bending. As a rule, it is possible slightly to flatten hardened cemented plates of moderate thickness by compression of the face, but bending, which puts the face in tension, almost always results in cracking. Any work that has to be done on the plate after the final heat is performed by grinding or disc cutting. Bending and tensile tests are usually carried out on specimens cut from the plate, and in the best practice some modification of Brinell's impression test, the Borba and Frémont drop test,

[13] Semp. "Local Annealing of Hard-Faced Armor Plates." American Institution of Electrical Engineers, 1896.

and vibratory tests on a duplex vibratory machine are also made.

The manufacture of cast armor is now in a highly efficient state. The metal is cast in a mold so formed as to produce a plate with a face having ridges and channels. The hot casting is removed to a special furnace, and allowed to cool slowly to prevent initial strains, after which the sand is removed and the casting carefully cleaned. It is then placed in a furnace, and upon the corrugated surface a carbonaceous composition is placed to a thickness of several inches, the furnace being raised to a temperature between 900 degrees and 1,100 degrees centigrade, which is maintained for several days. The plate is then allowed gradually to cool. The plate is again reheated and cooled four or five times, with a gradual reduction of the temperature to which it is reheated, and at the final heating the plate is dipped or sprayed with oil or water, according to the hardness required, so that the corrugated face, being hotter than the back, will become hard, while the back remains soft and tough.[14]

The fitting and erection of naval armor involves several operations of considerable technical interest. As an illustration of modern practice we may take a recent "first-class" battleship, the typical particulars of the armor and armament being as follows:[15]

ARMOR.

Belt, inches, ...	9
Deck, inches, ...	2—1
Side above belt, inches, ...	8—7
Bulkhead, inches, ...	12
Heavy-gun positions, inches, ...	12—6
Secondary-gun positions, inches, ...	7

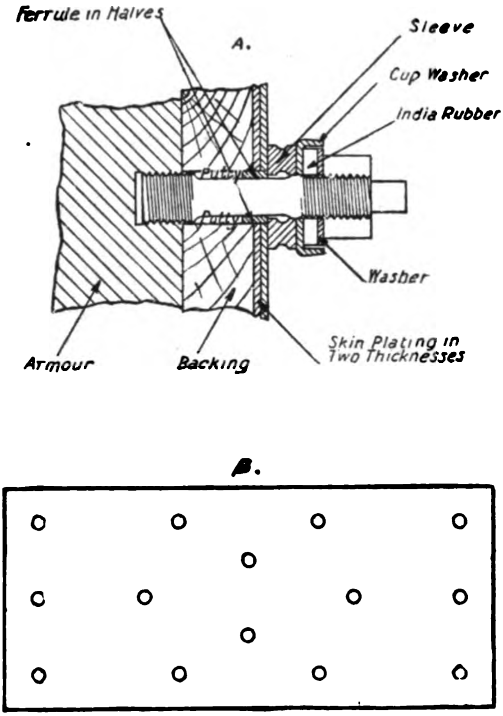
ARMAMENT.

Four 12-inch, four 9.2-inch, ten 6-inch, twenty-eight small guns.

The latest practice is to carry the side armor throughout the length of the ship, but to reduce the thickness to 2-inch

[14] Hadfield British Patent No. 17,174, 1905. "The Engineer," "Armor Plates," January 5th, 1906.
[15] "Brassey's Annual," 1905, page 235.

on bows and stern, this thin armor being riveted directly to plating and carrying no backing behind it. The armor bolts are of mild steel, Fig. 1, the number used being one bolt for



A = Section through armor bolt.
B = Disposition of bolts in plate, 14¼ feet by 6¼ feet.

Fig. 1.—ATTACHMENT OF ARMOR TO SHIPS.

every 7 square feet of plate. The usual dimensions, &c., of armor bolts are as follows:

Plate.	480 lbs.	400 lbs.	360 lbs.	320 lbs.	280 lbs.	240 lbs.	200 lbs.	160 lbs.
Diameter of shoulder of bolt at cup washer....................	4	$3\frac{3}{4}$	$3\frac{1}{2}$	$3\frac{1}{4}$	3	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$
Depth of hole in plate..	$3\frac{7}{8}$	$3\frac{9}{16}$	$3\frac{1}{4}$	$2\frac{15}{16}$	$2\frac{11}{16}$	$2\frac{5}{8}$	$2\frac{1}{16}$	$1\frac{3}{4}$
Depth of bolt in plate............	$3\frac{1}{4}$	3	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$

N.B.—The weight of one square foot of armor plate, 1-inch in thickness, is taken to be 40 pounds.

The bolts are made watertight where they pass through the skin plating, and are reduced in the center of their length to increase the resilience and to render them less liable to break under impact. The sleeves are fitted merely to fill up the length of the bolt before placing the nut on, and care is taken to get the lower row of side-armor bolts as near the lower edge as possible. The armor itself is not made watertight, and water finds its way between the armor and the backing.

The backing is East Indian teak, worked longitudinally on the ship's side. Behind the 360-pound armor it is 4 inches thick and tapers to about 2 inches behind the thinner armor.

The backing is worked in such thicknesses that the outer surface of the armor is flush, viz: 4 inches behind the 360-pound armor and 6 inches behind the 280-pound armor.

The plating is in two thicknesses of 17½-pound plate behind the 360-pound, 320-pound and 280-pound armor, and two thicknesses of 15-pound plate behind the 200-pound and 160-pound armor, each thickness serving as butt straps and edge strips to the other. In the most recent practice only one thickness of 30-pound plating is used, the butt straps and edge strips being worked on the outside of the ship. The edges and butts of the plating are planed and accurately fitted, being riveted with countersunk points. The plating is connected to the middle deck by a single angle bar 4½ by 4½ by 18 pounds, and to the main deck by a 4 by 4 by 13-pound angle.

The frames behind the heavy armor consist of two angle bars 10 by 3½ by 20-pound, and behind the thin armor the frames are 10 by 3½ by 24-pound down to 6 by 3 by 15-pound. The heavy frames are spaced about 2-foot centers, and the others at 3-foot centers. Two horizontal girders are worked between the frames intercostally. The plating is made watertight by calking the edges and butts, and the faying surfaces are coated with red-lead paint. When armor-bolt holes are cut out, the edges between the two plates are also calked, every effort being made to ensure the plating being watertight, irrespective of the armor and the backing. Red lead

is injected by means of a force pump between the backing and the plating. The outside of the backing and the inside of the armor is thickly coated with a stiff mixture of white and red lead before the plate is put on, and after the plate has been secured a thin mixture of the same materials is then injected.

Armor for coast-defence ordnance has quite a different character to that for naval purposes. Land-service shields usually have curved inclined fronts, and are supported by curved stays bolted to the sides of the carriage. Considering the small clearances allowed, these stays do not appear to possess adequate stiffness to prevent the shield setting back on impact and jamming the gears, particularly so when we remember that high-sighted guns are always under oblique attack, and that under these circumstances the work of impact is mainly the work of distortion or displacement of the shield. This aspect of the matter, with special reference to the box-like structure of naval casemates, seems only to have been considered in the French service. The shields contain openings for the gun and sights, and are either in one piece, or in two pieces let into one another at the front and bolted together by a thick plate on the inside.

ARMOR-PIERCING PROJECTILES.

Modern armor-piercing projectiles are made from a nickel-chrome ferro alloy, and are either cast or forged. Hadfield has given the following proportions of elements, other than iron, as being typical for air-hardened projectiles. Carbon, .8 per cent.; silica, .2 per cent.; sulphur, .04 per cent.; phosphorus, .04 per cent.; manganese, .12 per cent.; nickel, 2 per cent.; and chromium, 2 per cent.[16] The ingots for forged armor-piercing projectiles are usually made so that two projectiles can be obtained from one ingot, and the treatment of the ingot is somewhat similar to that adopted for the ordinary nickel-chrome plate. The ingots are not allowed to cool, but

[16] Hadfield. British Patent No. 6089. 1901.

are reheated and forged hydraulically. The forging is reduced roughly to finished size, the points being formed between swages, after which the projectiles are slowly reheated and annealed.[17] Armor-piercing shells have their cavities formed by punches and dies, and in the case of shells, such as the American, the base is closed by a plug screwed into the walls, which are not upset. In our service the base is continuous with the walls, and is closed by a base plug, the internal cavity being first punched or pressed, and the walls upset in a die when the ends only are hot. The cast-steel projectiles are cast point downwards, the body and dead head being in a sand mold, and the nose, or part of it, in a cast-iron chill. The mold is first dried and the chill coated with a carbonaceous composition, the core being carried on a spindle in the usual way. After casting, the shells are placed in an annealing oven, at about 1,400 degrees Fahrenheit, for a day or so, several castings being annealed at the same time. The shells are then machined, the dead head cut off and the cavity trimmed, the ogival head being either cut or ground to the correct radius. In our service the radius of the head is 2 calibers, corresponding to a nose angle of $2 \tan^{-1} \frac{\sqrt{7}}{3}$, or 83 degrees—12 minutes for all shot and shell. In America the 12-inch A.P. shot has a radius of head 24.18 inches, the 12-inch light shell 18.00 inches and the 12-inch heavy shell 18.075 inches.[18] A similar variation in practice obtains in other countries.

(In all probability this matter needs revision. The original series of experiments to determine the best shape of head for this class of projectile were carried out at the low striking velocity 1,500-foot seconds, against unbacked wrought-iron 14—12-inch plates, and unbacked 12—10-inch compound plates.[19] In each case the order of merit was assigned according to the depth of the indent made, a method which would be entirely unsuitable for modern hard-faced Krupp or

[17] Carnegie. "Armor-piercing Projectiles." "Proceedings," I. C. E., February, 1903.
[18] "The Engineer," "American Armor-piercing Projectiles," November 19th, 1897.
[19] Cunhill, "Experiments with Armor-piercing Projectiles," "Proceedings," R. A. Inst., Vol. XI, No. 7.

Terni plates. From the results of experiments on hardened steel plates by dropping heavily-loaded punches on them, I find the more highly elastic the plate the more blunt should be the nose, particularly for oblique impact. Furthermore, when the plate is defeated by what is generally called "boring action," I find the diameter of the punch relative to the thickness of the plate to be a deciding factor; whereas, when the plate is defeated by a smashing action, I find the angle of the nose and the diameter of the punch to be unimportant, the velocity rather than the weight appearing to decide the result; that is to say, a certain increase in the weight of the punch is less effective than the same percentage increase in the striking velocity.)

The head having been brought to the correct radius the base is then trimmed, bored and screwed, the groove for the driving band turned, and the projectile made ready to receive its cap. It is then brought to the correct weight, examined and hardened, after which it is ground to gauge, if necessary.

The method adopted for hardening projectiles varies in much the same way as the method for plates. Besides heat treatment, even cementation of the head has been carried out, the nose being afterwards tempered by cooling in oil. One method of hardening consists in heating the projectile, suspended point downwards, in a furnace, the point and shoulder being raised to a hardening temperature, whilst the point itself is protected from the direct action of the heat. The projectile is then removed to a hardening tank, and a cooling medium circulated under high pressure throughout the cavity, the projectile being afterwards completely immersed.[20] At Terre Noire the projectiles are oil hardened, the quality of the steel differing in different shells, being hardest in the largest calibers. The St. Chamond projectiles, generally of crucible steel, are heated to a cherry red, and gradually plunged into an oil bath until cold; but in some cases the projectile is plunged, for about eight or ten minutes, only up to the front band, after which it is completely immersed until

[20] Hadfield, American Patent, May 27th, 1898.

cold.[21] Armor-piercing projectiles manufactured at the Toubia Arsenal used to be heated in an upright gas furnace to a temperature between 780 degrees and 800 degrees centigrade; they were then transferred to an iron tank, and a jet of oil under a pressure of half an atmosphere acted on the point of the projectile until the tank was full, the capacity of the tank being sufficient to hold a weight of oil equal to ten times the weight of the projectiles. The method subsequently adopted for quick-firing armor-piercing projectiles was to heat them to about 740 degrees centigrade, and then plunge them into a bath of oil at about 24 degrees centigrade. They were then hardened again at the point, a jet of water being played on the inside and outside. The caps were also hardened, the top being heated to a bright red heat and the cap then plunged into water.[22] Air hardening, as introduced by Hadfield, is effected by directing a cold blast on the inside and outside of the projectile, which is raised to a higher temperature than when cooled by oil or water.

To test for dangerous initial strains the shells are kept for about three months, if possible, and some of them, after having been immersed in hot water up to the driving band, are plunged into cold water at least 130 degrees Fahrenheit lower than the temperature of the hot water. In America, before acceptance for ballistic tests, the projectiles are cooled to about 40 degrees Fahrenheit, and then suddenly plunged into a bath of water at from 180 degrees to 212 degrees Fahrenheit. When thoroughly heated, each projectile is plunged, with its axis horizontal, half-way into a bath of water at a temperature not greater than 40 degrees Fahrenheit, and after a brief period the projectile is turned over for a like immersion of the opposite side. The projectile is then removed from the bath, and tested under a hydraulic pressure of 500 pounds per square inch.[23]

With regard to the design of armor-piercing projectiles,

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[21] Nazro, "Modern Mechanism." Macmillan & Co., 1899.
[22] Cubillo, "Armor-piercing Projectiles" (Correspondence). "Proceedings," I. C. E., February, 1903.
[23] "The Engineer." "American-Armor-piercing Projectiles," November 19th, 1897.

the general practice is to put no restriction either on the material or the method of manufacture, the projectiles being accepted solely on the ballistic and other tests. The weight, form of head, maximum length, diameter over band, size and shape of driving band, are usually specified for armor-piercing shot; but shape, size, position of cavity and means of closing the base, are left to the manufacturer. In the case of armor-piercing shells the minimum capacity and minimum thickness of walls are also supplied.

A method by which the thickness of the base of a shell can be checked has recently been published, but it rests on the altogether artificial assumption that the base of the shell will tend to fail by shear along a cylindrical section parallel with the walls.[24] This, of course, the base will not do. The minimum thickness of the base to carry the powder pressure can be determined on the assumption that the base will behave as a plate, fixed at its edges, under a uniformly distributed load; then d being the diameter of the shot, p the maximum pressure on the base, and f the safe tensile stress, in the usual units, the thickness t is given by

$$t = \frac{d}{3}\sqrt{\frac{2p}{3f}}.$$

When the base is closed by a screwed plug it has to be ensured that the load on the plug can be carried by the shear strength of the thread. The usual thickness of base is considerably greater than that necessary to prevent fracture on firing; actual bases probably have a factor of safety of from 5 to 7.

The minimum thickness of the walls of an armor-piercing shell to prevent "setting-up" in the gun is much less than that necessary to prevent "setting-up" when striking a plate. The former can be calculated, but there is absolutely no theoretical guidance applicable to the determination of the latter. The minimum thickness of our armor-piercing shells closely follows a rule—

$$t = .144d + .084,$$

[24] Freeth. "Journal of the R. A.," December, 1905.

d being the caliber and t the minimum thickness in inches. American shells follow a rule—

$$t = .19d + .03,$$

and from the particulars available of French and German shells, the general practice appears to be to make t from one-fifth to one-seventh the caliber.

The simplest way to design a shell when the caliber, thickness of base, and minimum thickness of the walls has been decided, is to draw a section of the shell to a conveniently large scale, preferably on squared paper, and, assuming a probable shape of cavity, to determine graphically the weight of the shell, capacity of the cavity and position of center of gravity. A little experience enables one to adjust the shape of the cavity to bring the shell within the specified limits of weight, capacity, &c.

The factors affecting weight and proportion of projectiles do not generally permit of exact definition, but as a working rule we may take it that for service armor-piercing projectiles, the weight in pounds and the caliber in inches are connected by a rule, such as—

$$3\,w = d^{3.172}.$$

The tests of armor-piercing projectiles consist in firing them at certain velocities against a specified type of plate, the target being sometimes completed by a suitable thickness of backing and one or more skin plates. In the case of American armor-piercing shots, two are fired normally at a 1⅛-caliber plate, with velocity sufficient to carry them through the plate and backing. The acceptance of the lot depends on both shot getting through without cracks or deformation. An alternative test specifies that the shot must go through, whole or broken, 1 caliber of hard-faced plate at about 1,900 feet per second.[25]

CAPS.

About 1877, at Shoeburyness, a Palliser shot was fired at a steel-faced plate with a 2½-inch wrought-iron plate in front. It was found that the effect of the steel as regards the breaking

[25] "The Engineer." "American Armor-piercing Projectiles," November 19th, 1897.

up of the head of the projectile was completely neutralized, and its resistance consequently diminished to even less than that of ordinary soft-iron armor. English then suggested that a cap of wrought iron placed over the nose of the shot, so as to be cushioned between the shot and the plate on impact, would probably produce the same result; in accordance with this suggestion four capped 9-inch Palliser were fired in 1879. The cap used was similar to the present Russian type, and had a truncated nose; but the anticipated results were not realized. In the light of subsequent information it is not at all clear whether this failure was due to the insecure means of attachment, the low striking velocity—1,500 feet per second —or the already damaged plate against which the shots were tried. The cap fitted over the head of the projectile, and extended as far as the extractor holes, into which it was secured by screws. Above the point of the projectile the cap was 2¼ inches in thickness, and its point was truncated to a breadth of 2½ inches. It was found that the capped shot were in no way superior to the ordinary pattern, and no further experiments were carried out.[26] In 1894, the Russian Admiral Makaroff was firing capped shot against harder plates than the old compound, and with recorded success. The caps were reputed to be of steel, and were termed "magnetic," probably because of a magnetic means of attachment. Since that time experiments have proceeded in many countries, but not always with consistent success. Capped shot have been known to fail against plates at striking velocities suitable for uncapped shot to succeed. In August, 1894, two 6-inch chrome-steel Firth capped projectiles perforated a 6-inch Harveyized plate at about 1,800 feet per second;[27] but in July of the same year a Hadfield uncapped shot pierced a similar plate at 1,825 feet per second;[28] so that in this country, up to 1894, and against Harveyized plates, the cap had not demonstrated its value. In 1900, however, Hadfield was able to produce cap-

[26] Cunhill. "Experiments with Armor-piercing Projectiles." "Proceedings," R. A. I., Vol. XI, No. 7.
[27] "The Engineer," September 17th, 1897.
[28] Hadfield. "The Engineer," September 24th, 1897.

ped projectiles which, with a slightly higher velocity than the average usually employed, could readily perforate hard-faced Krupp plates.[29]

With steadier shot, improved manufacturing processes and materials, and at high striking velocities it has now come to be generally acknowledged that a cap adds from about $\frac{1}{4}$ to $\frac{1}{6}$ to the penetrative power against modern Krupp or Terni plates, at 1,800 feet per second. The cap, however, becomes rapidly less efficient as the angle of incidence is increased, so much so that beyond about 30 degrees the cap is useless. This fact needs to be insisted upon. It has been estimated that less than 10 per cent. of shots fired from a moving ship at sea will hit a target normally,[30] and considering normal impact with reference to the horizontal and vertical plane, it will easily be seen that at any range normal impact is altogether out of the question. With guns from high-sited batteries impact must always be very oblique.

There is a great variety of caps in use at the present time, and as great a variety in the means of their attachment to the projectile—Fig. 2. Hadfield's Heclon shell have the cap hammered into thumb-like recesses milled on the nose of the projectile. Johnson forces the cap into an annular groove with a few notches to prevent rotation. Firth sweats the cap to the nose with low-fusing-point solder, and also by running white metal into a groove in the cap and a corresponding groove on the nose. The Firth-Sterling caps are secured by small pieces of iron rod driven through holes in the cap into a groove on the nose; while many foreign examples are secured, like the experimental caps of English, by screws. Each of these means of attachment seems to fulfil the purpose for which it was designed, and whether the nose is grooved or not seems to be of little importance, since when shells break upon impact they do not generally do so through the groove as a weak section. The weak section seems to be determined by the initial stresses set up in the hardening process. In

[29] Hadfield. "Nature," January 25th, 1900.
[30] "Transactions" Institution of Naval Architects, Vol. XVIII, page 132.

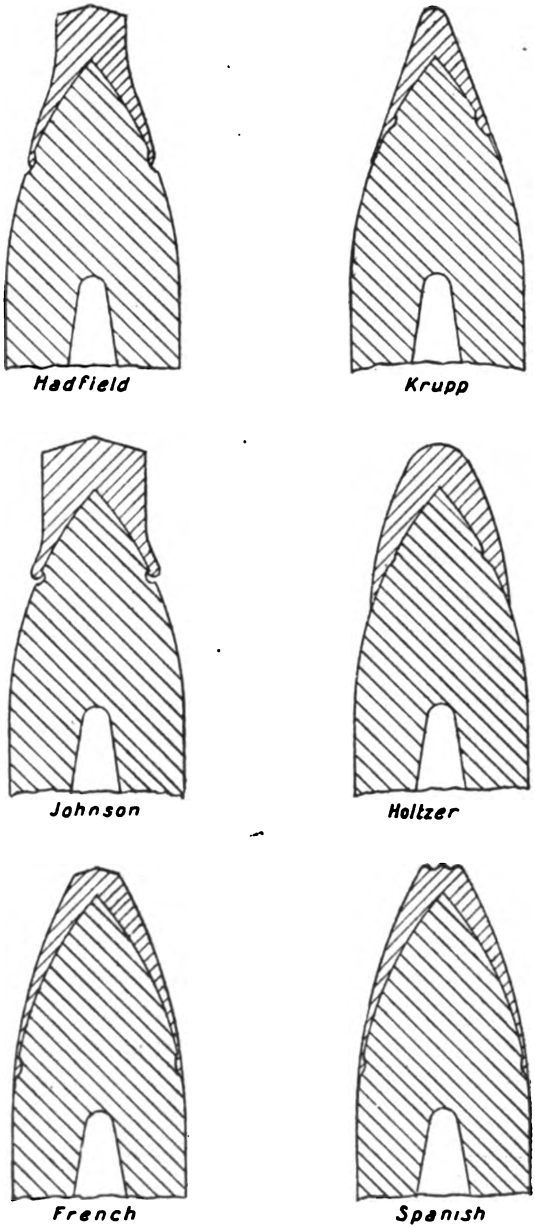


Fig. 2.—Capped Shot.

many cases the caps seem to have been designed to secure something patentable, and if all the caps are equally efficient as regards penetration, it would seem that a cap such as the Krupp would be the best, as involving less shortening of range by increased air resistance. This latter aspect of the matter is of some importance.

Many explanations have been proposed to account for the phenomena connected with the use of caps, and putting aside the unthinkable suggestion that the cap acts as a lubricant, the principal notions appear to be as follows:

(1) "The action of a cap was that when the projectile struck the plate it dished it at the point of impact, and perhaps strained it nearly up to its elastic limit. The first thing to touch the plate was the cap; the point of the projectile had then to travel practically the length of the cap before it touched the plate, and when the point touched the plate the latter was probably already strained beyond its elastic limit. That being so, the projectile had easy work in piercing the very hard surface of the plate; once the point had got in it was supported; and it only remained for the high energy contained in the projectile to do the necessary work of completing the perforation, if that energy was sufficient.[31]"

(2) "The hard face of an armor plate is designed to initiate the destruction of the delicate point of a projectile before the latter has obtained any appreciable penetration at all, for directly it has entered, as much as ⅛ inch even, it obtains a side support which increases the difficulty of breaking it, and the further it goes in the less support it needs, and the more it gets. It follows from this (1) that the hard face has only a very minute fraction of a second of time in which to perform its main function, and (2) that anything which will enable the extreme point of the shot to hold together during this brief instant is likely to save the projectile from the fatal initial pulverization, and to defeat the main object of hardening the face of the plate. This is the whole *raison d'être* of a cap.[32]"

[31] Dawson. "Armor-piercing Projectiles." "Proceedings" Inst. C. E., February 10th, 1903.
[32] Tresidder. "Brassey's Annual," 1905, page 347. Ewing. British Association, 1906. Discussion on "Armor Plates," by Edwards.

(3) "The causes of the rupture of the ogival head, and the means of preventing it, require delicate examination. Let us consider the case of the rupture of a projectile of good quality. . . . The point of the head was flattened by the blow, and took the form of a punch, which latter commenced the penetration; the ogival head afterwards entered the metal like a wedge. In proportion as the ogival head penetrated, the portion enclosed in the plate was violently compressed, whilst the part which remained outside underwent no pressure. This sudden passage from one state to the other, following the line of contact with the exterior surface of the plate, creates a critical state for such a rigid metal as chrome-steel; if this separation propagates itself into the interior rupture will take place. A remedy has been sought in the use of a more or less flexible and ductile envelope, which, placed upon the projectile, would transmit the pressure to it, and at the same time spread it out in a slightly graduated manner. This covering, or cap, of soft steel thus fulfils, from the point of view of the distribution of the pressures, an analogous part to that which is played, from the point of view of the distribution of the metal, by the curve which unites the two portions, of different diameters, of an axle, for the purpose of preventing the formation of a crack in the angle which the curves do away with.[33]"

Regarding these explanations it may be noticed that the first assumes that a soft-iron projectile fired against a face-hardened plate can, in the earliest stages of impact, strain the face of the plate up to its elastic limit, so that presumably in the immediate neighborhood of the point of impact the face is in a plastic or semi-plastic state. Now for the lowest striking velocity at which a cap is known to be effective, this assumption would not be generally admitted. And, further, if the cap strains the face in this manner the necessity for a very hard nose to the projectile is not at all clear; it seems reasonable to suppose that an ordinary steel nose would be

[33] Bertin. "Hardened Plates and Broken Projectiles." "Transactions" Institution of Naval Architects, July 6th, 1897.

sufficient to fracture already plastic material. Moreover, phenomena almost identical with that associated with caps can be produced by placing a layer of soft iron, or even copper or lead, in front of the plate instead of on the projectile; and evidently in this case the action can hardly be one of producing initial plasticity of the hard face.

The second explanation is typical of the one current in text-books. As far as I can understand its terms, it refers to the great resistance offered by materials to volumetric strain. That portion of the nose going through the cap is under lateral pressure arising from the resistance offered by the cap to the wedge action of the projectile. It is also under end pressure, at the extreme point, the measure of the initial resistance of the plate, and at the other end the pressure corresponding to the force of retardation acting on the remainder of the projectile. The front few inches of the nose is thus under a sort of fluid stress, and the only probable strain is that due to a change of volume—in other words, a crushing strain against which the material can offer very great resistance. The nose thus retains its shape, and is able to produce the intense local strain on the face which is favorable to fracture of the plate. But this explanation fails to account for the general experience that caps add nothing to the penetrative power below a striking velocity of about 1,700 feet to 1,800 feet per second, although this velocity is above that necessary to pulverize the bare point of the projectile. It also throws no light on the behavior of small capped projectiles, which, generally speaking, seem to gain little by the use of a cap, or on the behavior of capped shell against plates differing considerably from one caliber in thickness. M. Bertin's explanation obviously refers to the fracture of the head of the projectile rather than to the pulverization of its point; and the question at issue is not a consideration of the cause of fracture after the point has entered the plate, but how the point is able to enter at all and preserve its original shape.

The principal objection to these explanations appears to be

that they attempt to account for the phenomena resulting from a very complex play of forces, as if one out of many considerations affected the whole result. It has to be remembered that in the head of an armor-piercing projectile, where in a small mass we have a large variation of cross-section, initial stresses of considerable magnitude exist; and anything which can reduce the time rate at which the velocity of the projectile is reduced in the stage immediate on impact, is favorable to the instability of the material of the nose remaining undisturbed. The cap would, from this point of view, act as a cushion, reducing the maximum value of the retarding resistance set up by the face of the plate. In the next place, the cap acts in a manner favorable to the nose-resisting fracture, by transmitting the strain energy from the small mass at the point to the larger mass of the body of the projectile. As far as I understand Bertin, this is the main feature of the action of a cap. Now, it is known that at the short ranges usually adopted at proof butts, projectiles are not generally moving in a steady condition; that it to say, they have angular motion about an axis inclined to the axis of figure. As a consequence, armor-piercing projectiles at proof nearly always strike the plate with an incidence of a few degrees, and this oblique action, being accompanied by cross strain on the nose, is most favorable to the initial fracture of an unprotected point. Whether the pulverizing of the broken point is produced directly on impact, or whether it results subsequently from the crushing action of the remainder of the projectile, is not known. The cap, in being the means of transmitting this lateral strain from the point to the body of the head, favors the head remaining intact. Steel caps would obviously be better than soft-iron caps if their action has anything to do with the transmission of strain energy from the point to the body of the head.

The experience at our disposal lends support to the idea that the problem of armor-piercing projectiles is the problem of finding a suitable shape to the nose.

IMPACT.

The general problem of impact is one of great complexity, particularly so when the immediate aspect is the strain effect on the colliding bodies, and not the translational and rotational motions which the collision may ultimately produce. Some quantitative statements founded on more or less exact elastic theory have recently been furnished,[34] but they throw little light on the armor problem, because here we are concerned with a final state of fracture and not with a final condition of strain. And if it were otherwise, we should still have considerable difficulty in the approximate application of the ordinary elastic theory to a modern armor plate, which, being by no means isotropic, presents a marked variation in physical properties throughout its thickness. It is hence not a matter of surprise that the theory of armor under attack is confined to quantitative statements derived from empirical rules, which, with one unjustifiable exception, are candidly without theoretical basis.[35] These rules are confined to the solution of two classes of practical problems—the determination of the thickness of a plate just perforatable by a certain projectile, striking normally at a certain velocity; and the comparison of the merits of two plates, according to their behavior against the same nature of projectile. The first class is solved by reference to one of the many rules that a maker finds best conforms to his own practice, and the second by the introduction of a ballistic factor of merit, used in conjunction with a formula, such as that of Tresidder, believed to express the behavior of homogeneous unbacked wrought-iron plates; or a formula such as that of Jacob de Marre, for the homogeneous steel plates made by Barba at Le Creusot. When we remember the complex and difficult treatment of large armor plates, and the secrecy preserved with regard to the processes, and composition of materials, &c., it will be apparent that considerable diversity is to be expected in the published results regarding armor and projectiles. If in

[34] Smith. "Impact, Direct and Oblique." "The Engineer," March 10th, 1906.
[35] Tresidder. "The Engineer," July 24th, 1896.

nothing else, this diversity is most clearly shown in the great variety of empirical rules that are in use. It appears to be generally assumed that the thickness of a hard-faced plate just perforated, under normal impact, can be expressed as a single term involving functions of the weight, striking velocity and diameter of the projectile; and the changes are rung in every conceivable way, although it is well known that considerations other than these are effective in deciding whether the plate is to be defeated.

Krupp makes . . . . $t^2 \propto \frac{w\,v^2}{d}$;

Davis makes . . . . $t^{1.5} \propto \frac{w}{d} v^2$;

Tresidder makes . . . . $t^2 \propto \frac{w}{d} v^3$;

Vickers (Russian) makes . . . . . $t^{.7} \propto \frac{w^{.5}}{d^{.75}} v$;

t, being the thickness of hard-faced plate just perforated by a projectile of weight w, diameter d, and with striking velocity v. In fact, given a table representing fair average values of t and v as obtained from actual practice, then one has only to assume the form of any of the variables t, w, v and d, to make a very large number of these rules accurate to about 5 per cent., which is well within the limit of experimental error. I have plotted the above formula and taken a mean curve; assuming w and d to be connected by the relation already given, I find for service uncapped armor-piercing projectiles—

$$t = \frac{d^{\frac{5}{4}}\ v^{\frac{7}{6}}}{12{,}000},$$

and for capped projectiles—

$$t = \frac{d^{\frac{5}{4}}\ v^{\frac{7}{6}}}{10{,}000}.$$

Major Edwards, R. A., on the assumption that

$$\frac{w}{d^3} = .46$$

has pointed out that the formula for Krupp steel can be reduced to the form—

$$t=\frac{D\,v}{2{,}200},$$

for uncapped projectiles, and that a rough rule for capped projectiles is

$$t=\frac{v\,d}{1{,}950}.$$

There are several considerations which limit the utility of these rules. In the first place, they are only applicable to complete perforation by unbroken projectiles that have inappreciable remaining velocity after passing through the plate, and as most of the information on which the above formulæ have been built up has been obtained from tests in which the diameter of the shot was not widely different from the thickness of the plate, these rules should only be considered to have application to those cases in which the ratio $\frac{t}{d}$ approximates to unity. Against hard-faced plates the rules cannot be used for partial penetration. Thus, a 9.2 uncapped shot which will perforate an 8-inch Krupp plate at about 1,900 feet per second must not be expected to enter 8-inch in a 12-inch plate at the same velocity. Considerations of stiffness and thickness of cemented face affect the result, as will be indicated later. And again, in the trials, to cover the results of which the above rules were devised, several factors of importance were generally unknown. The striking velocity was usually calculated from the charge, and not experimentally observed. No cards were taken near the plate to ascertain whether, at the short proof ranges the initial unsteadiness of the shot had been damped by air friction, in spite of the fact that shots frequently hit the plate at points considerably out of the line of sight.[36]

In cases where the plate was perforated the effect of the backing, or what the remaining velocity would have been

[36] Cranz and Koch. "Untersuchungen über die Vibration des Gewehrlaufs." (München, 1899.)

without backing, was not determined; and in cases where complete perforation of the plate did not take place, what increase in striking velocity would have brought about perforation; although such experience with hard-faced plates as we possess goes to show that almost up to the point of yielding the plates show little or no indication of the approach to yielding.[37] Bertin, in his elegant discussion, assumed the rough rule, that the resistance of the backing was equivalent to an increased thickness of the plate equal to one-tenth the thickness of the backing.[38] The spring of the target and the permanent distortion of the plate were usually not recorded, or simply mentioned as an incident of the test. In addition, we must allow that plates from the same maker would show at least 5 per cent. variation under the ordinary tensile and impact tests, and hence we ought not to expect any of the above rules to show a greater consistency than such as lies within a 10 per cent. limit of variation. The behavior of modern armor is hence not rationally expressed by the usual tables, in which thicknesses of plate, just perforable, are given to two significant figures.

The most important practical problem connected with the attack of armor is the comparison of the qualities of defence of different plates. Two methods of meeting this problem are in common use. The first proceeds by a determination of the ratio between the thickness of a standard wrought-iron or steel plate, and the thickness of the hard-faced plate, that is just perforated by a certain projectile with a certain velocity. The second, by a determination of the ratio between the striking velocities necessary with the same projectile, in order to perforate the same thickness of the different plates. The comparison of thicknesses is the method adopted in this country, the standard plate being unbacked wrought iron, as represented by Tresidder's formula,

$$t^2 = \frac{w\, v^3}{d} \times \frac{1}{\log 8.8410}.$$

[37] White. "Transactions" Institution of Naval Architects, July 6th, 1897.
[38] Bertin. "Transactions" Institution of Naval Architects, July 6th, 1897.

On the Continent the comparison of striking velocities finds favor, the standard plate being soft iron, as represented by the Gâvre's formula, or more generally Le Creusot steel, as represented by the de Marre formula, which in French units may be written:

$$t^{1.4} = \left(\frac{1}{1{,}530^2} \cdot \frac{wv^2}{d^{1.5}}\right).$$

Neither of these methods can be considered as offering a scientific treatment of the subject, although they have come to have extensive practical adoption. Nevertheless, they make direct appeal to the naval architect and to the gunner. A comparison of defensive qualities measured in terms of thickness, and hence, for a given area of protection, in terms of weight, furnishes an immediate account of the minimum weight of armor to be carried to protect flotation, or for any other purpose, such as protecting gun bases, funnel bases, uptakes, &c., that may be thought desirable; whereas a comparison of these qualities in terms of striking-velocities furnishes an account as to ranges beyond which certain types or thicknesses of armor cannot be defeated by a specified nature of attack.

To facilitate the comparison of plates according to the English system, Tresidder's formula for unbacked wrought iron is generally taken as a basis, and used in conjunction with a ballistic term called the "factor of merit." This factor is defined as the ratio, R, between the thickness of plates, one of wrought iron and the other of hardened steel, both of which are just perforated by a given projectile with a given striking velocity. Another term, called the "factor of penetration," is due to Wolley-Dod. It defines the ratio between the thickness of wrought iron just perforated by a given round and the thickness of the plate against which the round is fired.

It is not at all clear what sort of comparative value we are to consider this factor of merit to possess; whether, in fact, we are to consider the quality of a plate as entirely furnished

by its factor of merit. Thus, supposing as a result of improved processes the factor of merit of a certain thickness of plate against a certain projectile went from 2 to 3, are we to consider that the improved plate offers half as much resistance again (resistance being understood to mean work spent in penetration) as the original plate? If so, a comparison of the plates by the continental method of velocity ratios should presumably show a like increase. But practical experience shows that there is no simple relation between the increase in quality, as measured by the ratio of thicknesses, and that measured by the ratio of velocities. Furthermore, the figure of merit is modified by the ratio between the thickness of the plate and the diameter of the projectile, in quite a different way from that of the de Marre coefficient. Hamilton has recently endeavored to insert considerations of this ratio in the ordinary Tresidder formula, but the data on which his formula are based are very restricted. He deduces,[39]

$$F = 2\left\{ 1 + .0014 \left(13 - d\right)^2 \left(\frac{t}{d} - 1\right) \right\},$$

F being the figure of merit, t the thickness of the plate, and d the diameter of the projectile. Introduced into Tresidder's formula for wrought iron, Hamilton gives the following formula for K.C. plates by capped A.P. projectiles, under normal impact:

$$\left(\frac{t}{d}\right)^2 F^2 = \frac{W}{d^3} v^3 \log^{-1}\left(0.55694 - 10\right).$$

The general phenomena arising from the impact of a shot on an armor plate appear to be as follows: As soon as the projectile touches the plate it suddenly experiences enormously increased resistance, and its rate of retardation is suddenly increased. The measure of the retarding force, in a direction parallel to that of motion, is directly proportional to the time rate of destruction of momentum of the projectile at the instant considered. This force acting on the nose of the projectile, in strictly normal impact, results in a wave of

[39] Hamilton. "Perforation Formulas." "Journal" United States Artillery, Vol. XXV, No. 3.

compression traveling from the point of the projectile towards its base, the velocity of this wave being directly proportional to the modulus of elasticity, and inversely proportional to the density. At the instant when this wave first travels towards the base, the rear portion of the projectile is moving towards the plate at the same velocity that it had immediately before impact, so that shortly after impact the nose and the base are moving in opposite directions, and if the thrust corresponding to this motion cannot be carried by the walls of the shell the walls crack or set up.

The pressure between the projectile and the plate results in a similar state of compression spreading throughout the plate, and this wavelike mobility of the particles of a plate under severe impact is visible in many plates.[40] Some very perfect specimens of these curves have also been obtained from firings at official proof butts. The velocity with which these waves are propagated also depends on the elasticity and density of the plate.

The immediate effect of such a rapidly-applied force is, as we have seen, both on the projectile and plate, to cause intense local strain. The magnitude of the strain energy per unit volume can be calculated when we know the stress and strain throughout a small elementary mass. If f is the average compressive stress, and E the modulus of the material for shock loads, then the strain energy per unit volume is measured by $\frac{f^2}{2E}$. If the materials at any time become stressed to the point at which permanent set takes place, $\frac{f^2}{2E}$ measures what is called the "resilience." We thus see that the amount of strain energy per cubic inch that the projectile or the plate can possess without permanent deformation or fracture is perfectly definite, and would be ascertainable if the physical constants under the actual conditions of the applied loads could be determined. If, then, at any time stress accumulates so rapidly that the strain energy exceeds the resilience of the

[40] Hartmann. "Comptes-rendus de l'Académie des Sciences de Paris," March, 1894. Institute of Naval Architects, July 6th, 1897.

materials, semi-plastic yielding or fracture must ensue. On the other hand, if by employing suitable materials, or by their suitable arrangement, we can ensure the strain energy being dissipated so fast from the point of impact that its value at any place is less than the resilience, local set or fracture will have been prevented. It follows as an immediate consequence that the fracture of a body under a blow depends on the ability of the body to dissipate strain in such a way that the strain energy is always less than the resilience.

We have seen that strain energy is measured by the product of stress and strain, and since the proportionality of stress and strain continues much beyond the ordinary static elastic limit,[41] we may say that strain energy, as far as we are here concerned, is measured by the square of either stress or strain, and the possibility of fracture is represented by the square root of the strain energy per unit volume. If the material is brittle, there is a sort of molecular instability which causes fracture at one place to extend to all neighboring places. Such an effect was produced in the old compound plates, the steel face of which frequently cracked into many pieces, like a piece of glass. The same feature is often seen in modern Krupp plates where the cementation has been inefficiently performed, the face breaking up with the well-known conchoidal fracture characteristic of obsidian. In these cases a moderate supply of energy, delivered with great rapidity, is sufficient to produce extensive fracture; but as the material gets less and less brittle we require a larger amount of energy delivered to get fracture throughout the same volume. It is for this reason that in cutting wood we use a wooden mallet and a long wooden-headed chisel. The mallet and chisel here act as a reservoir of energy, and the energy of the blow is slowly delivered by the edge of the chisel to the wood, but the rate is sufficient to cause fracture immediately in front of that edge. If the wood became more rigid without increasing in strength, so that the energy delivered by the blow would

[41] Hopkinson. "Effects of Momentary Stresses in Metals." "Proceedings" Royal Society, February 16th, 1905.

be transmitted more rapidly to other parts of the material, then we should have to use a more rigid mallet and a more rigid chisel. In dealing with a metal such as steel, we have to ensure a still greater rate at which energy can be delivered by using a short rigid steel chisel and a steel hammer.

It will thus be seen that if the wave of compression passing along the projectile and through the plate gives rise to such an accumulation of strain that the strain energy is greater than the resilience, fracture of the projectile and penetration of the plate must take place. We thus see the immediate function of a hard face to the plate and a hard nose to the projectile, hardness being used to signify, not mere capability of being scratched by other materials, but a high modulus of elasticity. If the point of the projectile is of highly elastic material, such as tempered steel, the strain energy of compression is rapidly propagated to the body of the projectile, and the yielding of the point is prevented. Precisely similar conditions obtain with regard to the plate. A highly elastic face enables strain energy to be rapidly conveyed from the immediate locality of impact to other parts of the body of the plate, and the strain energy not being allowed to accumulate to the limit of resilence, fracture is prevented. Since the velocity with which waves travel from the point of impact is determined by the physical properties of the material of the plate, it follows that any projectile, however soft, can fracture a plate, however hard, if only it has sufficient striking velocity; for the force of impact can be increased indefinitely with the velocity, whereas the rate of propagation of strain is limited by the materials used. But with actual projectiles and actual plates any increase in striking velocity is in favor of the projectile breaking on impact, and a broken projectile is all in favor of the plate not being defeated.

Again, in general, the rapidity with which a body can propagate strain depends on the direction of the blow with regard to the configuration of the body. Thus, a steel rod can be much more easily broken by a transverse blow than by a longitudinal blow, because the rate at which strain energy

is conveyed from the point of impact to other parts of the rod is less when the direction of the blow is at right angles to the axis of the rod than when it is parallel to that axis. This fact explains the rapid diminution in the power of a projectile to perforate a plate at oblique impact. The blow on the nose can be imagined resolved into an axial blow and a transverse blow. The strain energy due to the transverse component is not propagated as fast as the strain energy due to the axial component, and fracture of the nose is favored. The converse holds with the plate. Striking a plate at an angle is equivalent to sending component compression waves at right angles to its face, and along its face; and these latter waves, traveling at a greater rate than the former, tend to distribute the strain energy throughout a large volume, and thus prevent fracture, as the plate behaves like a rod, being more easily broken by a blow at right angles to its length than in the direction of its length.

It would thus appear that the usual methods of dealing with oblique impact are not rational; the problem is much more complex than the usual assumptions would lead us to believe. Treating the obliquely presented plate as one normally presented, but thicker in the inverse ratio to the cosine of the angle of obliquity, is a method for which we can find no theoretical sanction. Tresidder has proposed the formula,[43]

$$v = \cos \frac{\delta}{2} \sqrt[3]{\frac{C^2 t^2 D \times \log^{-1} 8.8410}{W \cos^2 \theta}},$$

which we may take to be an empirical rule awaiting trial, according to which the figure of merit of the oblique thickness is taken, and the mean direction of the hole in the plate to be halfway between the direction of impact and the normal to the plate. According to some trials of 6-inch capped A.P. projectiles by the Bethlehem Steel Company, quoted by Hamilton, the return to the normal was greatly increased by the use of a cap, and he proposes the following formula, in

[43] Tresidder. "Brassey's Annual," 1905, page 345.

which the return to the normal is taken as two-thirds the angle of obliquity:

$$\log P_\theta + \log d^3 + 9.443 = \log W + \log V_\theta^3,$$

in which P_θ is found with $\left(\frac{t}{d}\sec\theta\right)$ as argument, and

$$V_\theta = V \sec \frac{2}{3}\theta.$$

The method by which only the normal component of the striking velocity is considered as effective in producing penetration is obviously incorrect, since the transverse component has a most important effect in putting cross strain on the nose of the projectile, a condition most favorable for the fracture of the nose, as well as setting the face of the plate, at the point of impact, into a vibratory state, and thus relieving the face at the point of impact.

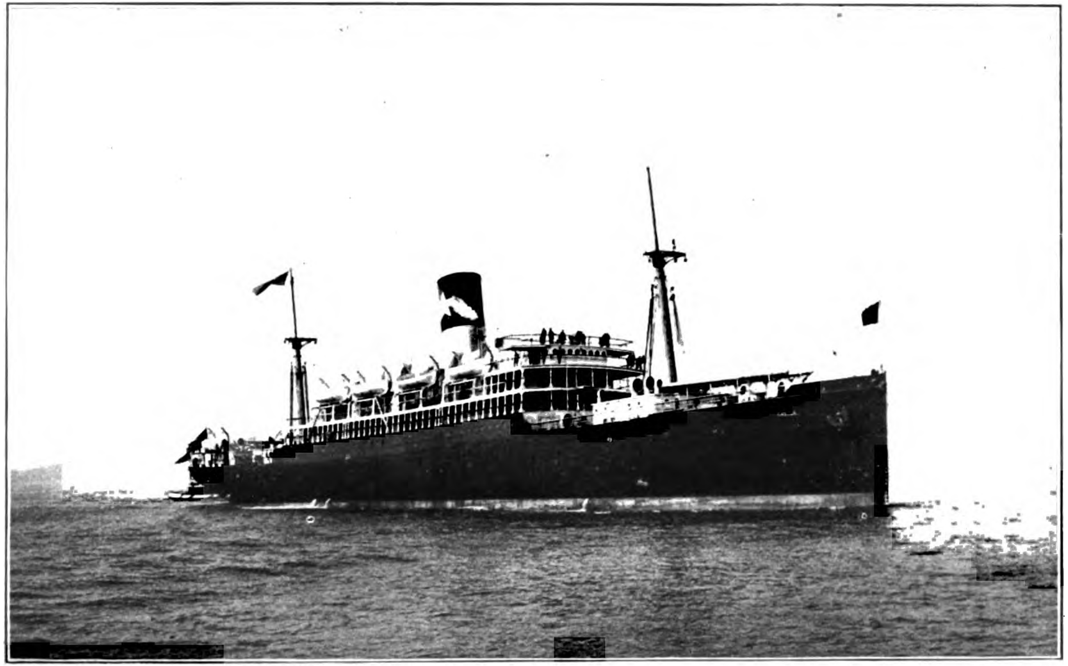
With regard to the action of a projectile, a somewhat artificial distinction has been made between a blunt and a pointed projectile. Thus Orde-Browne has it—"A plate may be perforated by 'boring' and 'punching.' An armor piercer, unless prevented, drives its sharp point through a plate at the point of impact, opening the metal outwards, and so clearing a path for the shot's body to follow. This process may be called 'boring' a hole. A blunt projectile cannot do this. It can only perforate by driving a disc of plate out in front of it. This is the way that a common shell with a plain head and a nose fuse perforates, and it may be termed 'punching,' and is resisted by the plate's tenacity and toughness.[43]" To an engineer this distinction must appear somewhat unreal. We invariably associate boring action with rotation, and in this respect projectiles, whatever be the shape of the nose, are similar. Even if the terms are considered to make reference to the character of the holes, they still seem to be inapplicable, since a punch will make a clean round hole in a plate if the latter is not brittle. The action appears to be one of degree,

[43] Orde-Browne. "Brassey's Annual." "Armor." 1899.

and not one of kind, as the usual reference implies. A pointed projectile is favorable to an initial localized strain, and hence to early fracture of the skin of the plate, which, owing to the molecular instability before mentioned, allows the fracture rapidly to extend. A flat-headed projectile acting in the early stages over a comparatively large area meets with a greater total resistance, but the resistance is less intense, and the corresponding strain is also less intense. Considering the time average value of the resistance, we see that in the case of the pointed projectile the portion of the plate just touching the nose is fractured in the very earliest stages of impact, before the waves of compression have had time to travel very far from that point, whereas with a flat-headed projectile fracture of the plate occurs at a later period, and the waves of compression have had time to travel from the area of impact to other portions of the plate, which are also fractured if the strain energy per cubic inch produced by the compression wave is equal to the resilience of the material.

The appearance of the hole made in the various plates presents some general characteristics. With the old iron or soft steel plates, a large bulge was first made at the back, followed by a tear extending from about the center of the bulge by three or four cracks to the edge of the hole, the edges being usually turned back. With hard, homogeneous steel plates the effect was quite different, and was much the same as that experienced with Gruson chilled cast-iron cupolas. The fracture seemed in no way to depend on the diameter of the shot, due, no doubt, to the reason already given—the rapid spread of the fracture with brittle metals. In the case of a modern Krupp or Terni plate quite another effect is met with. Elastic deformations of the plate may obtain a considerable value, and frequently the diameter of the hole is less than the diameter of the projectile. A large, slightly conical disc is punched out, such as would be produced by a considerable elastic deformation, accompanied by a tearing of the hole at its edges. Capped projectiles usually produce holes slightly larger than the projectile, and without surface scaling. At

oblique impact, if the plate is one just matched by the projectile at normal impact, scoops a few inches in depth are usually made, but if the plate is overmatched the capped projectile makes its characteristic hole, the general direction of which is inclined to the plate at a smaller angle than the angle of incidence.



By courtesy of "The Navy."

SOUTHERN PACIFIC S. S. "CREOLE," FITTED WITH CURTIS ENGINES.

BUILDERS' TRIALS OF CURTIS TURBINE STEAMER *CREOLE*.

BY CHAS. B. EDWARDS, ASSOCIATE.

The *Creole* is a first-class combined passenger and freight steamer built by the Fore River Shipbuilding Company for the Southern Pacific R. R. Co. She is fitted with twin-screw Curtis turbines, and the builders' guarantee calls, for a speed of 16 knots at 10,000 tons displacement. She can carry 420 passengers of all classes and 3,000 tons of freight. Figure 1 is a photograph of the vessel.

HULL.

The following are the principal dimensions of the vessel:

Length over all, feet	440
Length between perpendiculars, feet	416
Breadth, molded, feet	53
Depth, molded to upper deck, feet	37
Draught at 10,000 tons displacement, feet and inches	24–10
Coefficient of fineness, block	.645

BOILERS.

The vessel is fitted with ten Babcock & Wilcox water-tube boilers with superheaters, using natural draft. There is one smoke pipe 94 feet 4 inches in height and 153 square feet area. The total capacity of the ten boilers is:

Grate surface, square feet	770
Heating surface, square feet	28,500
Superheating surface, square feet	4,350
Ratio grate to heating	37
Working pressure, pounds	250

TURBINES.

The vessel is fitted with two 120-inch diameter, seven-stage Curtis marine reversible turbines, rated at 4,000 brake horse-power each, or 8,000 H.P. aggregate. Figure 2 is a view of the engine room looking aft, and Figure 3 shows one of the turbines with the wheels ready to assemble in the shop.

PRINCIPLE OF OPERATION.

The Curtis turbine is of the impulse type, and, in order to reduce the economical speed of rotation to a point suitable for direct driving of marine propellers, the turbine is divided into several pressure stages, and each pressure stage is provided with several rows of revolving buckets. The simplest possible impulse turbine would consist of a single wheel having a single row of buckets on which a jet of steam is directed by a suitable nozzle. Examples of such a turbine are the DeLaval steam turbine and the Pelton water wheel.

A simple single-wheel impulse turbine has its maximum economy when the buckets move with a velocity of approximately half that of the jet of steam. A jet of high-pressure steam, in a proper nozzle, will attain a velocity of about 4,000 feet per second when discharging into a vacuum, which would require the enormous speed of 2,000 feet per second for the buckets to obtain the maximum economy. This speed would require such a high number of revolutions that it would render direct connection to a propeller impossible, and would also produce centrifugal forces too great to be properly handled.

By using several wheels on the same shaft, each one utilizing the exhaust from the one preceding, the pressure drop and corresponding steam-jet velocity to be handled by each wheel is greatly lessened. Thus by using eight wheels, each in a separate chamber with its own nozzle, and so proportioning the steam pressure that the available energy is equally divided, the steam-jet velocity is reduced to about 1,400 feet per second and the bucket speed to 700 feet per second. This,

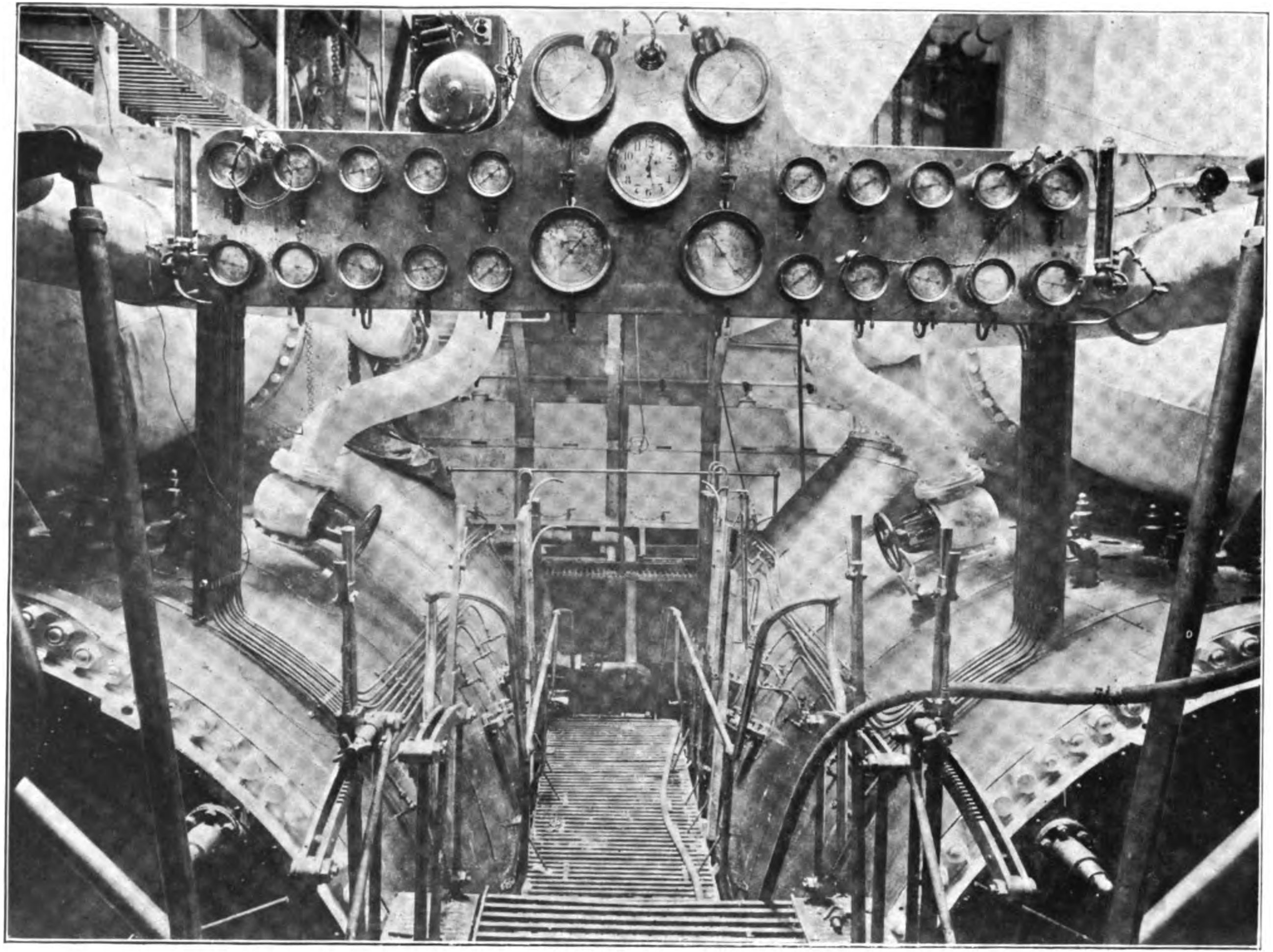


Fig. 2.—VIEW IN ENGINE ROOM, LOOKING AFT.

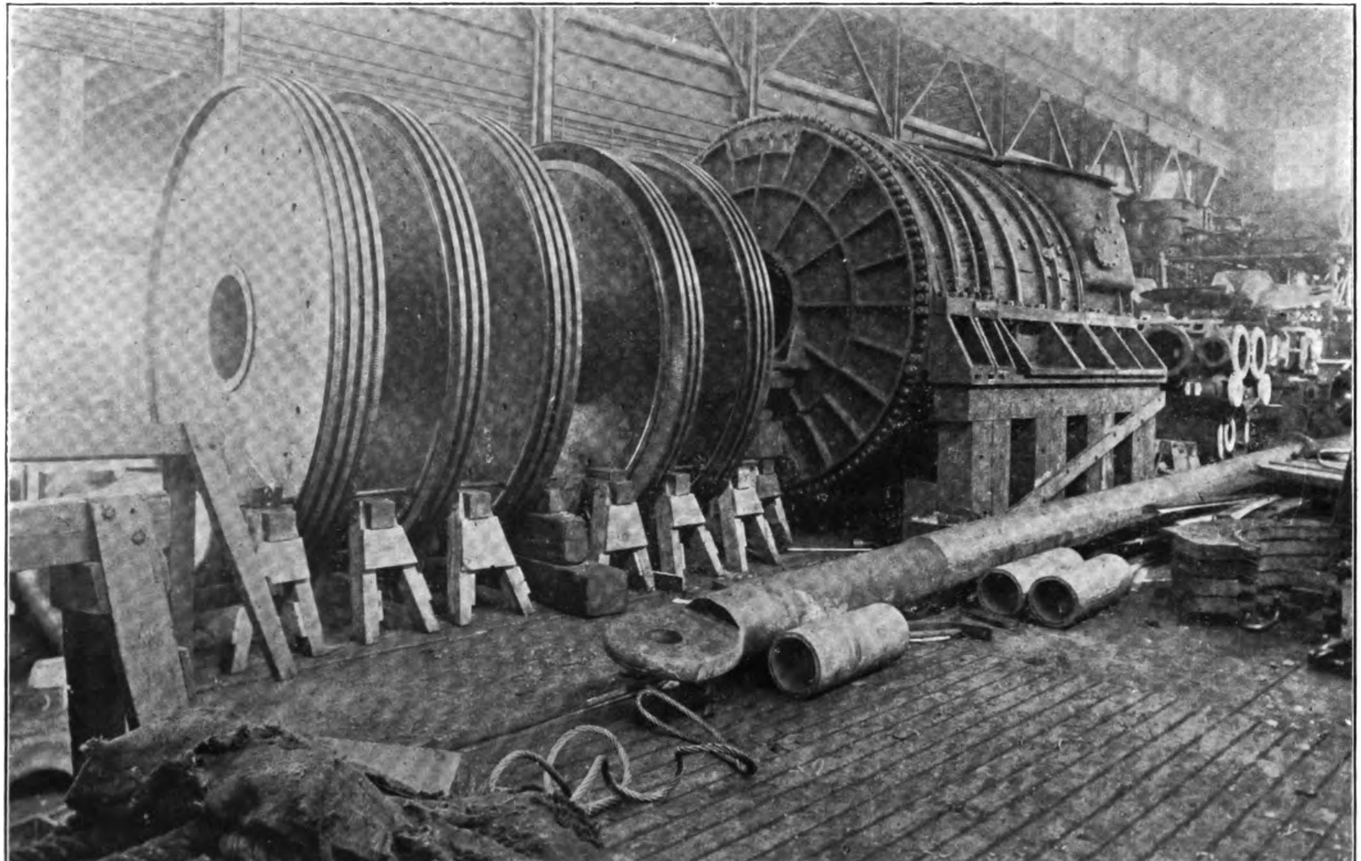


Fig. 3.—Turbine Casing with Wheels Ready to Assemble.

however, is still too high for satisfactory marine work, and, in order to lower it, each wheel is provided with three rows of moving buckets instead of only one. Each row of moving buckets will take out from the jet velocity approximately twice its own speed, so that the use of three rows on each of eight wheels will give a bucket speed of 230 feet per second. This speed can be used satisfactorily on high-speed vessels, but as the economy curve is quite flat at the point of maximum efficiency, the speed can be somewhat reduced without greatly reducing the economy, giving a fair working speed of about 160 feet per second. The above described method of reducing the economical bucket speed by means of several separate stages, each having several rows of buckets, is illustrated in diagrammatic form in Figure 4, which shows the first two

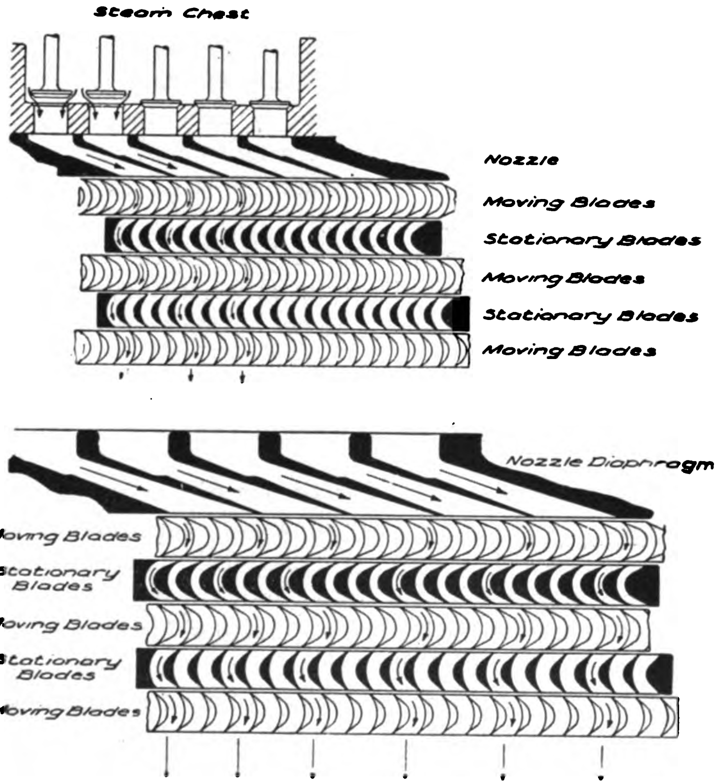


Fig. 4.—DIAGRAM OF NOZZLES AND BUCKETS IN CTRTIS STEAM TURBINE.

stages of a turbine of which the remaining stages would be similar. It is seen that between each row of moving blades there is a row of stationary blades, the object of which is to reverse the direction of flow of the steam as it leaves one row of moving blades and direct it into the next row of moving blades. The arrows show the direction of the steam as it passes through the nozzle and then through the moving and stationary blades.

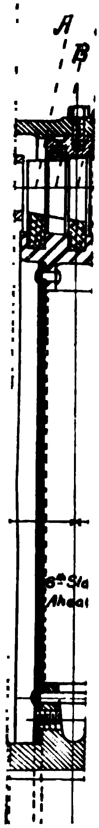
When the steam passes through a nozzle its pressure drops and it gains velocity; this velocity is absorbed by the moving buckets, thus converting energy of the steam into mechanical work. As the steam passes from stage to stage it drops in pressure, and when it passes from one row of buckets to the next in the same stage it loses velocity, but the pressure is the same in any one stage in all the bucket rows, and is also the same on both sides of the wheel.

This uniform pressure in all parts of stage obviates the necessity of small radial clearance between the ends of the blades and the casing, since there is no leakage around them. This clearance is, therefore, made as large as desired for proper mechanical construction.

The axial clearance, or distance between the edges of the blades, is also made quite large, its only limit being the slight disturbance of the jet which occurs at some distance from a nozzle or guide blade.

DESCRIPTION OF CURTIS TURBINES.

Figure 5 is a vertical section of one of the turbines as fitted to the *Creole*, and clearly shows the construction. The turbine consists of a cast-iron cylindrical casing divided by dished cast-iron diaphragms into a series of separate compartments (cast steel is used for two first stages, for strength). In each compartment or "stage" there is a separate wheel which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by self-


A
B
6th Sta
Aheat

aligning bearings as shown. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leak from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes which, prevent steam leaking out at the ahead end or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth-stage shell.

Cast-steel steam chests for ahead and astern running are attached to the front and back casing heads as shown, and are flanged for the main steam pipes. The nozzles for each stage are bolted to the diaphragms as shown, the diaphragms having steam-port openings cast in them to allow the steam to pass through to the nozzles.

Maneuvering is accomplished by means of two lever-operated balanced throttle valves, each taking steam from the main steam pipe, and one delivering to the ahead steam chest and the other to the astern steam chest. There are seven ahead wheels and two reverse wheels. The reverse wheels are mounted in the after end of the casing, and under ordinary ahead running they are in a vacuum and therefore do not waste power by steam friction. They are the same as the ahead wheels, except the blades are reversed. To reverse when going ahead the ahead throttle valve is shut and the reverse throttle valve opened, which is easily and quickly accomplished by the operating levers of the two throttle valves.

In the ahead steam chest there are twelve valves, each opening one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Nine nozzles will give full power, leaving three for overload. The astern steam chest has the same number of nozzles, but only six have valves.

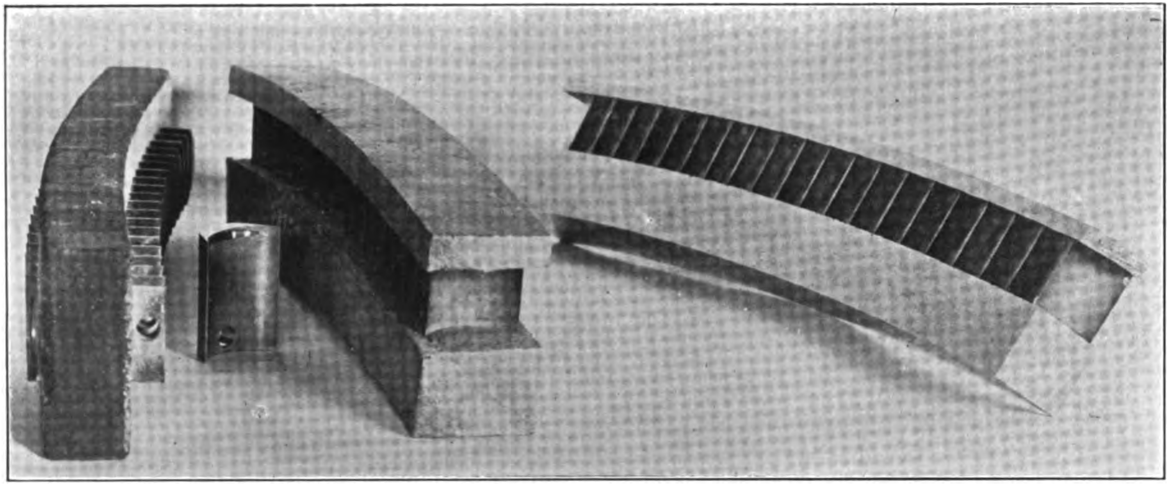
For maneuvering, the nozzle valves in the steam chests are left open and the speed is controlled by the throttle valves.

On each of the other six ahead stages two nozzle valves are fitted. Each valve closes one-fifth of the nozzle area of its stage. These valves are closed when running continuously at reduced speed, so that the proper steam pressure can be maintained in each stage.

Drain pipes are provided connecting each stage with the next, so the condensed steam in any stage will pass to the next one of lower pressure and there give up a part of its heat to do useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A regular marine thrust bearing is attached to the forward end of the turbine shaft. In addition to taking the propeller thrust this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is one-tenth of an inch on the first wheel and increases to one-quarter of an inch on the seventh wheel. The thrust is put at the forward end, so that any unequal expansion of the shaft and casing will be allowed for at the aft end where the clearance is largest. This axial clearance is very ample to allow for all unequal heat expansion that may occur and any mechanical irregularities and leave sufficient leeway for adjustment.

The construction of the blading is shown in Figure 6. The blades are made of special bronze accurately formed to the required shape by being extruded through a die and thus made into long bars. These are cut into lengths as required by each stage, and each length is notched or drilled through at the ends as shown at B. A number of blades are built up into a segment by casting on a composition base on the inner ends and a shroud on the outer ends as shown at C. In order to do this casting the blades are held in a sand core, with the ends projecting slightly as shown at A. After casting, the base and shroud are milled off true, thus completing the segment as shown at D.

The ends of the blades fuse into the cast parts, thus mak-



A B C D

By courtesy of "The Navy."

Fig. 6.—BLADING SEGMENT.

A Blades held in core previous to casting on base and shroud.
B Blade cut to length and notched.
C Segment after casting on base and shroud.
D Segment machined to size.

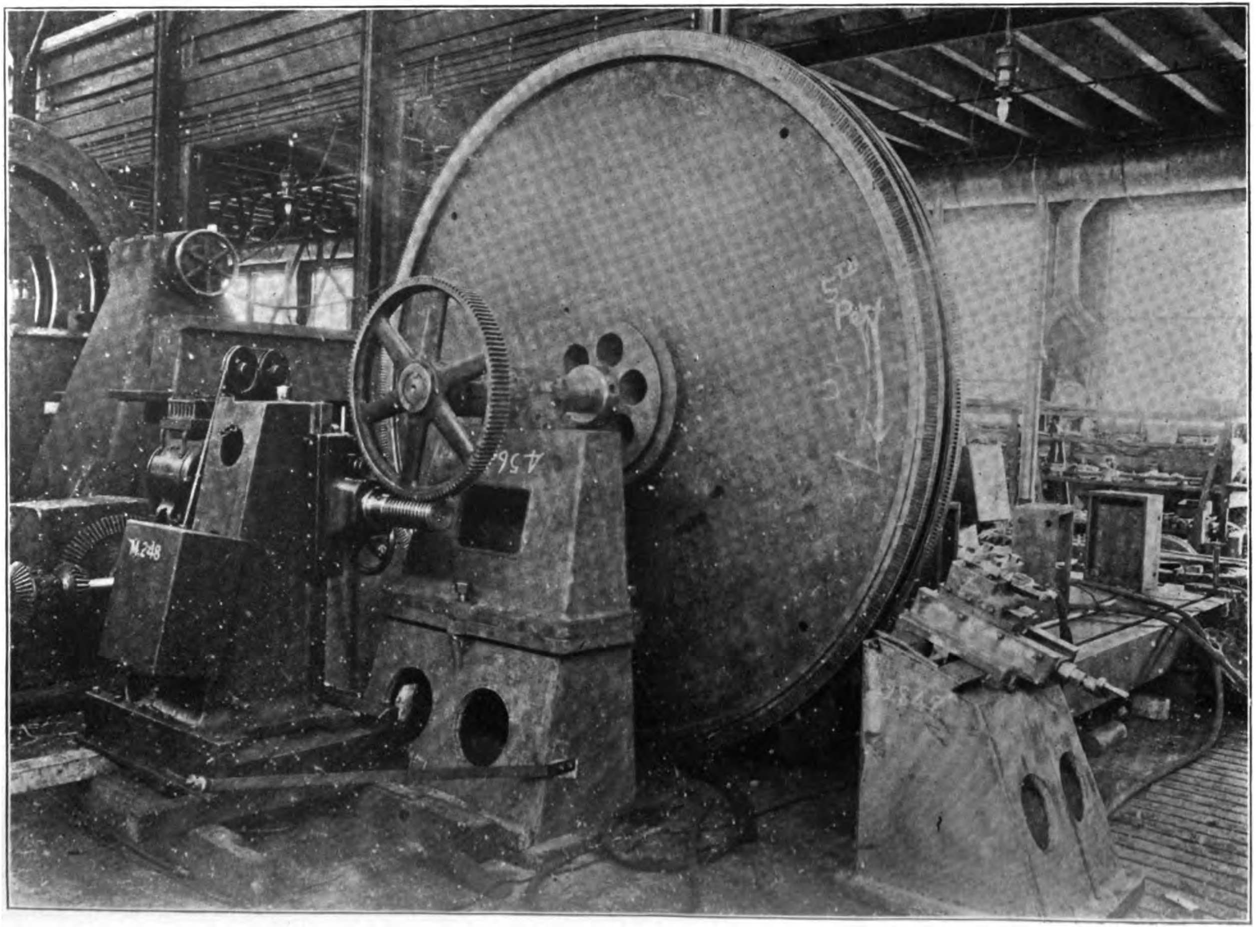


Fig. 7.- Calking in Blade Segments.

ing practically one solid piece of the entire segment. If any blade should not happen to fuse, the metal will flow into the notch or hole and effectually keep it in place. The extruding process gives the blades a hard, smooth skin, which reduces steam friction and effectually prevents wear and erosion of the blades.

The segments are held in the wheel rims by inserting the bases in rectangular grooves around the steel rims and calking the edges of the grooves. This calking is done by means of a pneumatic calking tool mounted on special adjustable pedestal, while the wheel is slowly revolved, as clearly shown in Figure 7. On the segments actually used in this vessel the shrouds were riveted on instead of being cast like the bases, but on the next turbines constructed they are cast as above described.

The bases and shrouds are overhung slightly beyond the edges of the buckets, so that if by any accident the wheels should be moved axially the blades would not come in contact and be damaged, but the overhanging bases and shrouds would protect them. These bases and shrouds are very strong, and can stand forcing together so hard that the turbine would be brought to a stop without any serious damage occurring. This very important feature renders stripping of the blades a practical impossibility in this design of turbine.

As before stated, the radial clearance at the ends of the blades can be made as large as desired without having any effect on the operation, and, as constructed, it varies from ¼ inch to 2 inches, depending upon the proximity of the stationary blade holders, as is clearly shown in Figure 5.

To allow for the increased volume of the steam as it expands in passing from stage to stage at lowering pressures, the length of the blades are increased, as shown in Figure 5, and also the arc of the nozzles is increased, thus giving greater area of passage in each succeeding stage. Also, in any *one* stage, the blade lengths are increased in each succeeding row, because the velocity falls as the steam passes from row to row, although it is at practically constant pressure throughout the stage.

In order to keep the pressure in the shell as low as possible, the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the firsts tage, and one-eighth in each of the other stages. This requires the first-stage nozzle to be of the expanding type, but all the other nozzles are of the parallel-flow type. Also the first-stage wheel is provided with four rows of buckets instead of three, as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions.

PROPELLERS.

On the first trial variable-pitch propellers were used, but the results were not satisfactory. These were changed to two true-screw propellers, each having the following dimensions:

Number of blades	3
Diameter, feet and inches	10–9
Pitch, feet and inches	8–2
Pitch ratio	.76
Projected area, square feet	33.78
Developed area, square feet	36.0
Disc area, square feet	90.1
Immersion of tip at 24 feet 10 inches draught, feet and inches	11–5½

MEASUREMENT OF TRIAL DATA.

The brake horsepower of the port turbine was measured by a Föttinger torsion meter, which was calibrated in the shop on the section of shafting on which it was used.

The steam flow of the port turbine was measured by two tanks which received the discharge of the port air pump alternately and which emptied into the port hotwell. These tanks were calibrated by weighing the amount of water which each held.

The temperature of the steam in the turbine stages and steam chests was measured by high-grade glass thermometers inserted in mercury cups let into the steam spaces. The other

temperatures were measured by the regular ship thermometers which were permanently installed.

Pressures in the three lower stages and in the condensers were measured by mercury columns. Other pressures were

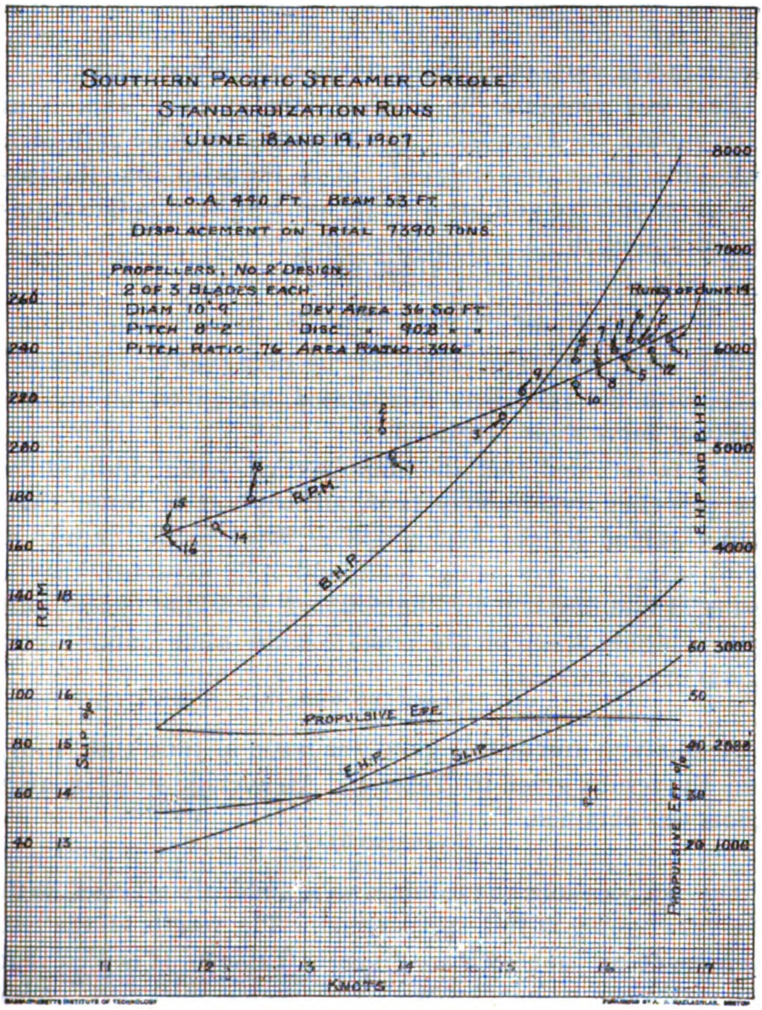


Fig. 8.

measured by the regular ship's gauges, which were kept adjusted for their working pressures by a weighted gauge tester.

Revolutions were taken from the ship's continuous counters.

The time on the measured mile was taken by two observers with stop watches, and upon crossing the line at both ends an electric signal was sent to the engine room to read the counters to obtain the revolutions on each mile run.

Torsion-meter and counter readings on the port turbine were read every minute, other engine-room readings every five minutes, and boiler observations every ten minutes.

Coal was measured by weighing the passing buckets as they entered the firerooms. On May 1st and June 19th each individual basket was weighed, but on the other runs the baskets were counted and the average net weight as before determined was used.

RESULTS.

STANDARDIZATION.

The standardization results of the runs over the measured mile, with the propellers finally used, are given on accompanying curve sheet, Figure 8.

It is to be noted that the propulsive efficiency obtained by the propeller used is very poor, although it is better than that obtained by the first propeller fitted to the vessel. A third design is now being built, and it is expected that a material improvement will be obtained, which, of course, will give a higher speed of the vessel for the same brake horsepower of the turbine.

FULL-SPEED RUN.

A four-hour run at full speed was made in June 19th, 1907, and the results given in Table 1. An average speed of 16.7 knots was obtained.

REVERSING TEST.

On two occasions the turbines were suddenly reversed with the vessel going at full speed ahead. The results of the test were made on June 18th are given in Table 2. It is seen that the vessel can be handled with remarkable quickness. Also she maneuvered perfectly when being taken through the drawbridge and in the narrow and crooked channel from Fore River. The vessel was docked at the Pier in North River without the assistance of tugs.

ECONOMY TESTS.

Steam consumption was measured on the port turbine on several runs, and the results are given in Table 3. During the run on May 17th the steam and vacuum conditions were maintained the best and most constant, and, therefore, the best water rate was obtained. The water rates obtained, namely, about 16 pounds per brake horsepower per hour for best steam and vacuum conditions, and about 17 pounds for average conditions, are about those to be expected of a triple-expansion reciprocating engine; suitable allowance being made for engine friction to compare these figures with *indicated horsepower* on which reciprocating-engine performances are based. In view of the fact that the power and speed of this vessel are low for her size, thus necessitating a bucket speed much below that giving maximum economy, these results are considered very satisfactory. The economy results also agree within a few per cent. with the performance of the experimental demonstration turbine from which the present design was developed, thus showing that results obtained on a small machine may be expected to be duplicated with confidence on a similar larger design.

By taking the power of the starboard turbine as equal to that of the port at the same revolutions, the total power of the vessel is obtained, and the economy results for both turbines on the run on May 17th are given in Table 4.

The results of boiler performance are given in Table 5.

Table No. 1.—Four-Hour Full-Speed Run.

Run was from Wood End Light around Boston Light Ship and return, on June 19th, 1907.

Down:	Distance by chart, knots	31.5
	Elapsed time, hours, minutes and seconds	1·52·20
	Average speed, knots	16.83
Back:	Distance by chart, knots	31.1
	Elapsed time, hours, minutes and seconds	1·52·25
	Average speed, knots	16.6
Total Run:	Average speed, knots	16.71

Table No. 2.—Backing Test.

Maximum power ahead = 247 R.P.M. 3.790 B.H.P. 7.17 Torsion meter for port turbine.

Time.	Chest steam pressure, gauge.	Vacuum in exhaust shell.	R.P.M.	Torsion meter reading.	B.H.P.	Per cent. of full torque.	Per cent. of full power.
1·10	...	...	...	4.0	...	55.8	...
1·11	200.	9	...	3.0	...	41.9	...
1·12	192.	9	135	2.95	854	41 2	22.5
1·13	187.	9	132	3.1	877	43.3	23.1
1·14	182.	13	127	3.05	870	42.6	22.9
1·15	175.	13	132	3.05	863	42.6	22.8
1·16	169.	14	136	2.95	860	41.2	22.7
1·17	165.	...	140	2.85	861	39.8	22.7
1·18	...	...	...	2.95	...	41.2	...

Time from throwing telegraphs till vessel came to a dead stop, minutes and seconds	2·42
Distance traveled, lengths	3½ to 4½
Time from throwing telegraph till reverse valves on turbines were opened, seconds, port	5
starboard	8
Time from opening reverse valve till turbines stopped, seconds, port	11
starboard	15

NOTE.—Low vacuum is caused by carbon packing in ahead and not being properly arranged to take reverse pressure.

Table No. 4.—Economy Results for Both Turbines for Run on May 17, 1907.

Item.	9·40 to 10·00	10·00 to 10·30	10·30 to 11 00	11·00 to 11·30	11·30 to 11·50
R.P.M., port	236.5	239.6	244 8	250.8	249.7
R.P.M , starboard	228.4	232.8	239.7	242.2	240.6
Brake horsepower, port	3,450	3,585	3,805	3,950	3,920
starboard, est	3,030	3,195	3,480	3,595	3,520
total	6,484	6,780	7,285	7,545	7,440
Steam flow per hour, port	58,950	59,260	62,200	63,500	63,200
starboard, est	58,400	58,700	61,500	62,850	62,550
auxillaries	10,600	10,600	10,600	10,600	10,600
total	127,950	128,560	134,350	136,950	136,350
Water rate, all purposes	19.75	18.95	18.45	18.16	18.35
Dry coal per B.H.P. per hour, all purposes	2 28	2.19	2.13	2.1	2.12
port turbine only	1.98	1.91	1.89	1.86	1.86

Len	10·00 to 10·30	10·30 to 1
Nu	7	[illegible]
Ste	238.7	23
	72.3	7
Ex	1.20	[illegible]
Av	354.3	34
R.I	221.2	23
To	5.56	[illegible]
Br	...	3,14
Ste	...	?60,00
W.	...	1
Ef	...	3
Ef d	...	4
Cr t	...	—[illegible]
W.	...	...

Table 5.—BOILER RESULTS.

Date	May 17.	June 19.
Duration	9·45 A. M. to 11·50 A. M.	8·30 A. M. to 12·30 P. M.
Boiler pressure, pounds, absolute	262.9	262.0
superheat, degrees Fahrenheit	82.0	56.0
Feed-water temperature, degrees Fahrenheit	192.0	186.0
B.T.U. in steam	1,259.0	1,242.0
Factor of evaporation	1.139	1.128
Water evaporated per hour, actual	132,800.0	132,964.0
from and at 212 degrees Fahrenheit	151,300.0	150,000.0
Wet coal burned per hour	15,800.0	15,361.0
Dry coal burned per hour	15,340.0	15,020.0
Combustible burned per hour	13,530.0	13,770.0
B.T.U. per pound of dry coal	13,880.0	14,653.0
Actual evaporation per pound of wet coal	8.40	8.67
dry coal	8.65	8.85
combustible	9.81	9.65
Evaporation from and at 212 degrees Fahrenheit, per pound of wet coal	9.58	9.67
dry coal	9.87	10.0
combustible	11.2	10.89
Boiler efficiency, per cent	68.5	65.7
Dry coal per square foot of grate area, per hour	19.9	19.5
Water evaporated per square foot boiler heating surface, actual, per hour	5.48	5.48
Water evaporated per square foot total heating surface, from and at 212 degrees Fahrenheit, per hour	5.3	5.25
Tons of coal burned per hour, actual	7.06	6.87

NOTES.

TAR OILS FOR DIESEL ENGINES.

When the Diesel engines were first introduced it was thought that the great future of the new motors rested on the fact that an engine had been brought out which would consume any kind of liquid fuel. As is often the case, the development took place on rather different lines. The Diesel engine showed its own preferences as regards fuel, just like the gasoline motor, and other difficulties had first to receive attention and be overcome. This fuel problem is not serious in our country. A Diesel motor is fed with so-called paraffin, which can be obtained at very cheap rates, and motors of this type are more commonly in use here than on the Continent, and in Germany, whence they hail, and where import duties keep the fuel question alive. A discussion of these features is, unfortunately, complicated by the disgraceful state of the nomenclature concerning oils. The engineer and the scientific chemist are sure to misunderstand one another in such a discussion, and it seems high time to put a stop to this confusion.

In Germany tar oils are broadly distinguished as lignite-tar oils and coal-tar oils. The tar oils gained from the distillation of wood and peat are not at present of much importance. Again, speaking broadly, lignite-tar oils are considered a suitable fuel for Diesel engines, while most of the coal-tar oils are regarded as unsuitable. The coal-tar oils that are sometimes utilized in Diesel motors are creosote oil and anthracene oil, whose boiling points lie above 240 degrees centigrade. These oils are very much cheaper than the lignite oils; they can be obtained in Germany for from 25s. to 50s. per 100 kilogrammes, while the lignite oils command prices ranging from

80s. to 110s. The low price of the former product is partly due to some undesirable qualities of these oils: bad smell, want of uniformity, &c. But as the German production of coal-tar oils is now double that of lignite oils, the utilization of the cheaper fuel in Diesel motors is a question of considerable interest. Of the lignite-tar oils the less expensive grades, which do not contain much paraffin proper—*i. e.*, hydrocarbons of the C_nH_{2n+2} series, valuable as wax—are generally used in Diesel motors; the density of those oils lies between 0.860 and 0.926.

Although there is no inducement to use a high-grade paraffin oil as fuel, general experience seems to have suggested that a tar oil containing more than 2.5 per cent. of paraffin would not answer in Diesel motors. Mr. Paul Rieppel, of the Vereinigte Maschinenfabrik Augsburg and Maschinenbau-Geselleshaft Nürnberg, found, however, that a higher percentage of paraffin did not in itself disqualify a tar oil as Diesel-motor fuel. He raised the paraffin proportion by steps to 15.2 per cent., and worked motors with such fuels for more than fifty hours without experiencing any trouble. As, moreover, crude petroleum and gas oils of foreign origin, containing considerably more than 2.5 per cent. of paraffin, answer perfectly in Diesel engines, the conclusion that the paraffin percentage was not, as such, a criterion for the suitability of a Diesel-motor fuel appeared to be justified. Mr. Rieppel thereupon resolved to investigate the whole problem with the object of ascertaining which oils and mixtures can be recommended and the cause of their suitability. The experiments have been carried out in the engineering laboratories of the firm at Augsburg and Nuremberg, in the chemical laboratory of the Gewerbe Museum at Nuremberg, in the engineering laboratory of the Technical High School at Charlottenburg, and in the Institut für Gährungsgewerbe (fermentation) in Berlin. Though the subject has not been exhausted, and investigation will be carried further, some important conclusions can already be drawn from Mr. Rieppel's account of his research in the "Zeitschrift des Vereins Deutscher Ingenieure."

We may also draw attention to a memoir on liquid fuels for internal-combustion motors, published by Mr. Kutzbach, a colleague of Mr. Rieppel, in previous numbers of the same journal.

As a rule, a 35-horsepower motor was worked under normal conditions in the thirty-eight series of experiments made on twenty-one oils and mixtures. All these materials were tested chemically and physically, and these latter tests were extended, for the sake of comparison, to some oils, of which sufficient quantities could not be obtained for long trials. The number of oils investigated thus comes up to twenty-six. One of the first materials tested was benzine. It proved very unsatisfactory. There were misfires and violent explosions, the exhaust would shoot out as a column of fire and soot, and pistons and valves became badly tarred. When the load was reduced below normal, the fuel would not ignite at all. Mixtures of benzine with tar-oils, especially the creosote and anthracene oil mentioned, worked better; at full load up to 40 per cent. of the latter products could be added to benzine; but such mixtures are not really to be recommended.

The calorific values of the oils were obtained with Mahler bombs. For the determination of the viscosity, Engler's method was first applied, and the experiments were conducted at 20 degrees centigrade (68 degrees Fahrenheit). As the viscosity changes with the temperature, however, it was thought best to measure the viscosity at the temperature at which the oil is forced through the atomizer. These temperatures could not be determined directly—not even with the aid of thermo-couples—and the following indirect method was, therefore, resorted to: A 12-horsepower Diesel motor was fitted so that the atomizer could be removed 45 and 42 seconds after stopping the motor, and immediately be dropped into a calorimeter. The temperatures thus deduced for the atomizer were 74 and 76 degrees centigrade. These temperatures are lower than might be expected, since the atomizer projects partly into the combustion space; but there is cool-

ing, directly through the oil and air injected, and indirectly through the circulating water; and a temperature of 80 degrees centigrade (176 degrees Fahrenheit) was, therefore, adopted as the probable temperature of the oil at the moment when its viscosity has effect. So far as the experiments go, the viscosity is not a general criterion. A certain viscosity value may be most favorable for a particular kind of oil, but this viscosity cannot guide us as to other oils. Flash point, temperature of ignition and calorific values were also determined, but none of these features seem to be guiding characteristics for the suitability of an oil.

The elementary analysis, the determination of carbon and hydrogen and of their ratios, seemed to be a better criterion. The lignite-tar oils are mostly hydrocarbons of the fatty series corresponding to the formulae $C_n H_{2n}$ or $C_n H_{2n+2}$. In the coal-tar oils the aromatic hydrocarbons, benzine (C_6H_6), and its derivatives, predominate, and they are relatively poorer in hydrogen than the oils of the former class. The combustion analyses, made in Dennstedt electric furnaces with the aid of platinized quartz, proved that the ratio H : C in the good Diesel-motor oils ranged from 1.53 to 2; while in the coal-tar oils the value of this ratio was generally below 1. The anthracene and creosote-tar oils previously mentioned gave the ratios 0.82 and 0.81, and were thus unsatisfactory; by mixing them with paraffin oil the ratio can be raised to 1.39, however, and they then become fairly suitable. The distillation tests proved still more characteristic than the ratio H : C. The distillation, performed more conveniently in copper stills than in glass retorts, indicates what portion of the oil is volatilized or gasified at a certain temperature. When this distillation was watched it was seen that some oils would generate oil gas at a fairly uniform rate, while others would gasify irregularly in its fits and starts. The former oils are suitable, the latter not. The lignite oils showed, on the whole, a more uniform distillation then the coal-tar oils, and one understands why the former should constitute the better fuel. The Scottish tar oil, obtained from cannel coal and boghead shale,

ranked with the lignite oils, the gasification being quite uniform. It differed from the lignites in so far as the lignites are completely gasified at a temperature of 400 degrees centigrade, while the Scotch fuel leaves a pitch residue representing 43 per cent. of the fuel bulk. The formation of a pitch does not appear to be injurious in a Diesel motor, however, as the subsequent combustion is sufficiently intense to burn all such solid residues.

It was further of interest to study the behavior of the various fuels at high pressures and temperatures. The spherical bombs which Mr. Rieppel constructed for the purpose of getting complete combustion records did not answer, and he adopted a simpler bomb of his colleague, Mr. K. Kutzbach. This bomb is cylindrical, with a disc flame, the stem being formed of two concentric cylinders; the annular space is charged with the oil, the thermometer inserted in the inner cylinder, and the bomb closed by screwing down the cover, to which the pressure gauge is attached, to its flange. The annular space has a capacity of 75 cubic centimeters, and it is charged with about 30 cubic centimeters of the oil, which is heated by Bunsen burners. Control observations made with steam and benzine charges demonstrated that the pressures and temperatures were not recorded quite correctly, but the accuracy appeared sufficient for comparisons. Plotting temperatures as abscissae, and pressures as ordinates, the experimenters obtained curves which would first rise regularly, and then ascend steeply, indicating that the pressure first increased with the temperature, and that afterwards the further supply of heat did not cause any further rise of temperature, but increased the pressure, owing to decomposition and the generation of gas. The gas is, according to Kutzbach, a mixture of hydrogen and hydrocarbons, and it is said to be the hydrogen which first catches fire, a statement which appears to be disputable. In some cases the thermometer actually fell a little on further heating; this fall, if real, and not due to instrumental errors, might be ascribed to a dissociation of the gas. Crude petroleum gave a regular curve, indicating evapo-

ration without decomposition. Creosote and anthracene tar oils gave similar curves up to a certain point; then a lively gas generation set in. With Scotch liquid fuel and the lignite oils, this gas generation sets in at comparatively low temperature, and that circumstance would explain the suitability of these oils for Diesel motors. With coal-tar oils the gas generation becomes brisk only at higher temperatures, and they would not give sufficient gas for spontaneous ignition in the Diesel motor.

The difference in chemical constitution may account for this different behavior. The coal-tar oils are rich in derivatives of the benzine series, whose benzine nucleus is not easily split up, and such a splitting up would have to precede decomposition. The lignite oils contain essentially hydrocarbons of the fatty series, whose side chains are more easily split up; liquid fuel naturally ranks with the lignites, being a shale oil. We also understand why a high percentage in hydrogen should be desirable, though the hydrogen percentage does not directly decide the suitability of the oil as Diesel-engine fuel. The paraffins of the fatty series are rich in hydrogen, and the hydrogen in the resulting gas mixture burns readily. These conclusions are supported by the tests on carburetting oil which A. Spiegel has quite recently published ("Schilling's Journal für Gas und Wasser," 1907, No. 3); an oil makes a good carburettor if it is rich in paraffins.

One point, however, appears to be rather contradictory. Benzine was characterized as unsuitable for Diesel motors, while it answers perfectly in internal-combustion motors, and, being a uniform substance, it should give, and did give, a regular curve in the bomb. The discrepancy is explained by the different working conditions. In the gas motor the benzine vapor is drawn in, compressed by the next stroke, mixed with air, and the explosive mixture is ignited by an electric spark. In the Diesel engine the liquid fuel is injected into the highly-compressed air, in which it is at once to catch fire spontaneously; the whole process is much quicker. That the Diesel engine will not work well with

benzine is no drawback, of course. If the mixture could locally be heated to higher temperatures, or otherwise be made to burn spontaneously, the Diesel engine might work with the less combustible coal-tar oils. Mr. Rieppel has made very satisfactory laboratory experiments in this direction, and fed Diesel motors with coal-tar oils for weeks. The matter, however, remains, so far, in the experimental stage.—"Engineering."

CORDITE.

All the nitro-powders, being highly exothermic chemical compounds, in place of a mere mixture of reagents, such as common black powder, are much more liable than the latter to deterioration when stored under high temperatures. The disastrous explosion on board the *Jéna* will be in the minds of all, and our own Navy and Army have also suffered from this unfortunate characteristic of the modern type of propellant, though, fortunately, on a much less serious scale. In December, 1899, a 6-inch cartridge on the *Revenge* ignited spontaneously, and two other cartridges in the same box were also found to have gone off, and last October a similar incident took place on board the *Fox* whilst at Bombay. In the autumn of last year, moreover, there were two spontaneous explosions at cordite magazines in India. When it recently became known that the Home Office had laid an embargo on certain cordite manufactured by Messrs. Kynoch because it contained an unauthorized ingredient in the shape of mercuric chloride, it was not therefore surprising that a large section of the public jumped to the conclusion that these disasters arose from defective ammunition purchased from the firm in question, and the politicians with socialistic leanings rejoiced in a new argument for the abolition of the private contractor. Unfortunately for their thesis, this ammunition turns out in each case to have originated from the Government factories at Waltham, and there is thus every prospect that the nation will continue to reap the benefit of the ingenuity and enterprize of private firms.

It is impossible for any one individual to know everything. It is, therefore, most important that the labors of the Government experts, able as these are, should be supplemented by the work of independent experimenters. Indeed, most of the radical improvements in armaments, from the Maxim gun to the submarine boat, have originated in this fashion. Cordite itself is no exception, for though it was first compounded in the Government laboratories, the authorities frankly admitted that their investigations were based on the pioneer work of Nobel. Again, until the introduction of the so-called Palma Trophy ammunition by the King's Norton Company, the Lee-Enfield rifle proved inferior at long ranges to the Mannlicher and Mauser, whilst subsequently it held the advantage. It is, therefore, a matter for congratulation that the spontaneously-ignited ammunition cannot be represented as the natural outcome of a contractor's "greed for dividends," but was actually manufactured at Waltham, under the superintendence of officials as to whose complete honesty there is absolutely no question.

Nevertheless, although our national forces have suffered no disaster from the ammunition supplied by Messrs. Kynoch, the action of this firm in adding mercuric chloride to the cordite supplied by them appears to be indefensible. In itself the addition does no harm, as the official experts admit, and it is quite possible that Messrs. Kynoch in all honesty believed that the addition was actually an improvement. In such circumstances it is not uncommon for contractors to take matters into their own hands, incorrect as this may be from the standpoint of law and ethics. It is, for example, notorious that certain specifications for some special gunmetals for steam fittings specifically exclude the use of lead, yet all practical men know that a certain proportion of this is necessary if a sound casting is to be obtained. A small quantity is, therefore, quietly added by some foundrymen; but it seems, in certain cases at any rate, to entirely disappear in the operation of casting, although, before vanishing, it has performed its desired function of causing the tin and copper

to form a satisfactory alloy. Where the interests concerned are, however, so large as in the present instance, no firm, no matter how well assured they may themselves be as to the resulting improvement of their product, have the right to violate a clause of the specification drawn up by the experts held responsible by the nation for the quality of the ammunition supplied to our sea and land services.

The objection to the presence of mercuric chloride in cordite has in some cases been misunderstood. It is not, as Mr. Haldane very clearly pointed out in his admirable speech on Tuesday evening, that good cordite is injured by the addition of this ingredient. Of this there appears to be no evidence, and Messrs. Kynoch, no doubt, believe that it is actually improved by it; but the fact remains that the presence of even a minute quantity of this compound will enable a bad and dangerous sample to successfully pass through the ordinary official "heat" test.

In this test 20 grains of cordite are placed in a test tube, in the upper half of which is suspended a strip of filter paper which has been impregnated with a mixture of starch and potassium iodide. The upper half of this strip is moistened with a 50 per cent. solution of glycerine in water. The tube and its contents are then heated up in a water bath to 180 degrees Fahrenheit, and at the end of a certain time, more or less prolonged, according to the quality of the sample, a brown line makes its appearance at the junction of the dry and moist halves of the paper. With a good sample of the regulation cordite this line should not appear till after the test has lasted twenty minutes. The brown coloration is due to the production of nitrous fumes, which liberate the iodine from the potassium iodide. The addition of alkalies, urea, or, above all, mercuric chloride to the sample renders the test invalid, the nitrous fumes liberated combining with the addition, so that they consequently fail to reach the potassium iodide. This fact has long been known and has been recognized in the specifications governing the acceptance of nitro-powders, not only of our own, but of other governments. The

regulations of the United States Army Ordnance Department, for instance, say that "the presence in a powder of mercuric chloride, or alkali, or any other substance which may in any way mask the heat test, will be sufficient to cause its rejection." The specifications of our own authorities are equally explicit, so that the action of Messrs. Kynoch in adding the mercury salt to the cordite supplied by them is, as already pointed out, quite indefensible. If persuaded that good cordite was made still better by the addition, their proper course was obviously to convince the authorities of this, and get them to amend the specification accordingly; but in adding without permission a reagent which, it is admitted, would enable dangerous samples of the explosive to pass the official test, they accepted, even on the most favorable interpretation of their action, a most serious and dangerous responsibility.

The authorities are evidently taking a sane and conservative course in the matter. Instead of destroying without investigation many tons of what may be perfectly good and safe cordite, they are carefully examining and testing the whole of the doubtful stock. Some extremists in the House have taken the view that the Government should be held immediately responsible for the consequences of any decision which may finally be come to in the matter. Politicians in opposition are peculiarly prone to this contention, but in technical and scientific matters the real responsibility must rest upon the experts of a department, and not on the mere political mouthpiece. The latter is only open to blame if, after procuring the best scientific advice available, he fails to follow it. In an analogous case, for instance, the responsibility for the success of a surgical operation must fall on the operator, and not on those who called him in.—"Engineering."

"WHITE-FORSTER" WATER-TUBE MARINE BOILERS.

The "White-Forster" patent water-tube marine boilers are well known and have been largely adopted, especially for torpedo-boat destroyers for the British and foreign navies, and also for other vessels. Machinery fitted with "White-Forster"

boilers, Fig. 1, shows the design of boiler as fitted to the destroyer H. M. S. *Ness*.

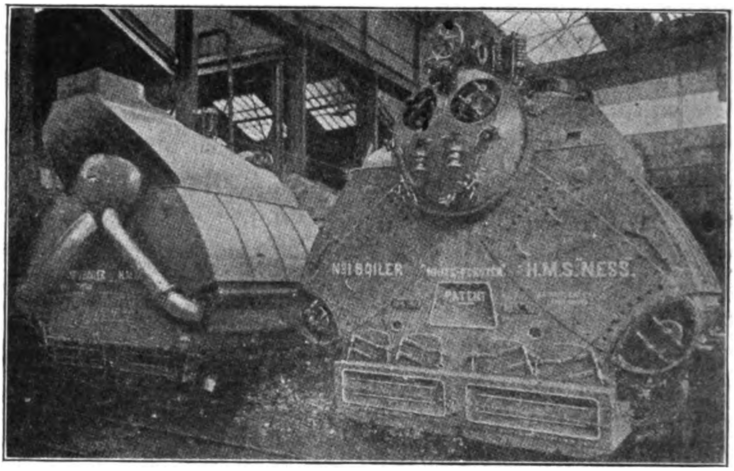


Fig. 1.

Several important improvements have recently been made, and the accompanying illustrations show the latest types. Fig. 2, shows a 2,000 I.H.P. boiler for H. M. S. *Cricket*, coastal destroyer.

The principal features of the " White-Forster" boiler will be readily understood from Figs. 1 and 2. From these it will be seen that there are two lower water drums connected by the generating tubes to an upper steam and water drum, the water level being arranged as to entirely drown all the tubes. The radius of curvature of each tube is the same, and the curvature is only sufficient to determine the direction of movement due to expansion, and also to facilitate cleaning and repairs. The tubes are arranged in position in a transverse section like staves in a section of a barrel, the continuation of the line of curvature of each tube, passing through the top drum in line with the end manhole, allows each tube to be inserted or withdrawn through this manhole, so that it is possible to withdraw any tube without disturbing the remaining tubes, and as every tube has the same curvature they can

be readily cleaned, internally, by a tube brush having a rigid handle curved to the same radius. The boiler and its casings are constructed entirely of mild steel, no castings of any kind being employed. Large downtake tubes are fitted. Ordinary manhole doors are fitted to all drums, studded doors being avoided. As the inclination of the generating tubes is considerable, and usually varies between 40 degrees and 60 degrees, the circulation is definite and rapid, keeping the tubes free from deposit. No special water baffles or separator plates are found necessary, the usual internal steam pipe being sufficient to give dry steam. Special care is taken in designing the casings and uptakes so as to make the boiler as efficient as possible. Patent baffles are fitted so as to divide the uptake into two or more parts, in such a manner that the gases are drawn equally over the whole tube surface, and at right angles to it, without offering any resistance or increas-

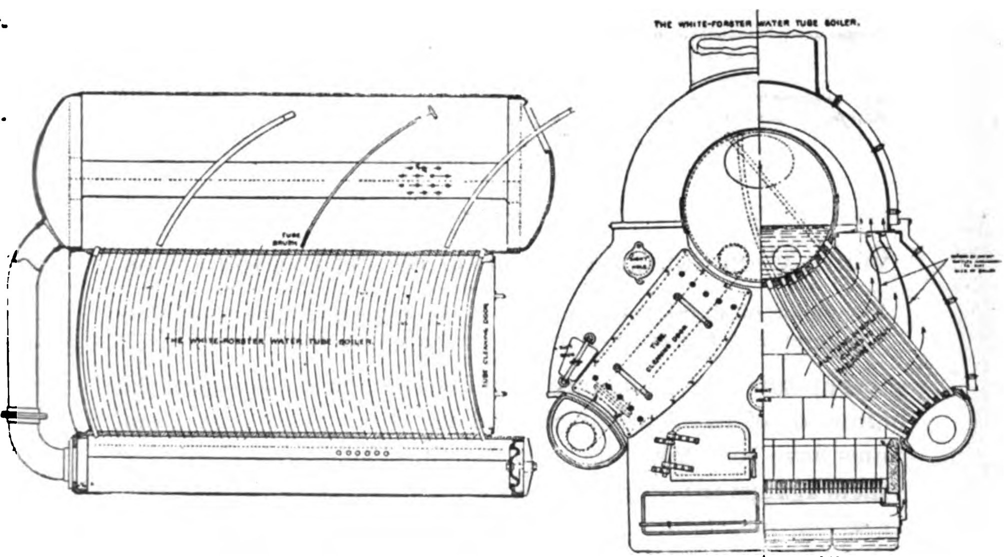


Fig. 2.

ing the air pressure. Air holes are arranged at the sides and ends for the admission of air above the grate ; sight holes are fitted for the inspection of the uptakes and combustion cham-

ber. Large doors are fitted at the front end, so that the whole length of tubes is exposed for cleaning, and a brush can be passed between the tubes from end to end of the boiler.

With regard to steam efficiency, the following table shows average evaporative results from a recent test of a "White-Forster" boiler with oil fuel:

Evaporation Trial, One Boiler.	*Full Power.*	*Low Power.*
Duration of trial, hours	2	2
Heating surface, square feet	4,000	4,000
Volume of combustion chamber, cubic feet	302	302
Mean Results, per hour.		
Water evaporated (actual), pounds	45,410	15,750
Oil burnt, pounds	3,800	1,145
Water evaporated per pound of oil fuel (actual), pounds	11.94	13.75
from and at 212° F., pounds	14.5	16.7
square foot of tube surface (actual), pounds	11.35	3 93
from and at 212° F., pounds.	13.61	4.75
Oil burnt per square foot of tube surface, pounds	0.950	0.2865
combustion space, pounds	12.6	4.82
Steam pressure, pounds per square inch	205 to 212	202 to 207
Oil pressure, pounds per square inch	210 to 220	100
Air pressure in stokehold, inches of water	4.6 to 4.9	0.95 to 1.0
Oil temperature at burners, degrees Fahrenheit	200 to 215	180 to 200
Temperature of atmosphere, degrees Fahrenheit	77	80
stokehold, degrees Fahrenheit	88 max.	90 to 97
Smoke	Very light to nil.	
Number of burners in use	*11	†4
Total for two hours, Water evaporated (actual), pounds	90,820	31,500
Oil burnt, pounds	7,600	2,290

* Open ½ a turn. † Open ¾ of a turn.

VENTILATION AND REFRIGERATION OF AMMUNITION HOLDS.

Translation of abstract of paper read at the Bordeaux International Congress in Naval Architecture, June 27, 1907, by ADRIAN BOCHET.

The safety of ammunition holds has always engaged the attention of naval architects, and for many years past measures have been taken for isolating these holds, for protecting them against the various causes which may lead to an accident, and for inundating them in the event of danger. Modern powders, and the considerable development in the use of machinery on board ship, have increased the risks in a large proportion. Modern powders have excellent ballistic properties, but are also most unstable, and their instability increases very rapidly with an increase in temperature. By their grad-

ual alteration they also set free a quantity of inflammable gases which may give rise to explosive mixtures. The development in engines and boilers, in auxiliary engines, and in the extent of steam pipes laid throughout the ship, had led to a greater heating of the various compartments. The distribution of the armament, and the necessity of providing ammunition holds in proximity to the guns to be served, have often resulted in the holds being in locations that are particularly unsuitable from the point of view of temperature. Therefore, concurrently with a gradual increase in sensibility to heat, which forms one of the characteristics of modern powders, causes tending to augment the temperature of the ammunition have increased in a large proportion also, and numerous attempts have been made with a view to inure the artificial refrigeration of the holds.

Cooling by ventilation alone is a simple and safe means to prevent the accumulation of explosive gases, but is quite insufficient to ensure a decrease in the temperature of the holds when the temperature of the air outside reaches 20 degrees (say 70 degrees Fahrenheit), and when the causes making for an increase of temperature in the holds attain a certain importance. The heating of the air which enters the ship occurs rapidly on contact with the warm bulkheads, by reason of its low specific heat. A rise of 10 degrees centigrade has often been noted for courses which appeared as short and direct as possible between the upper works of the ship and the fans placed in the holds. Cooling by the means resorted to in refrigeration holds, such as those for the transport of meat, appeared at first to constitute a final solution of the problem. But this failed completely, owing to the very great difference existing between these holds and those for storing ammunition. In the former there is no ventilation, and they remain closed during the whole passage. They are also maintained at a very low temperature, and one result of this is that the small proportion of moisture contained in the local air on starting gets deposited on the products contained in the refrigerating chamber, and does not injure them in any way.

Ammunition holds, on the other hand, require ventilating. Even were one so imprudent as to do away with ventilation completely, or to reduce it in too large a measure, the requirements of the service would demand the frequent opening of the hold, thus allowing every time the outside air to penetrate into it. Finally, the temperature which it is suitable to maintain in an ammunition hold is much above 0 centigrade (32 degrees Fahrenheit). One result of this frequent renewal of the air in an ammunition hold is that the moisture it contains is condensed on all the surfaces which are maintained at a lower degree of temperature than that of the outside air, and the water which cannot be congealed, as in the case of a provision hold, trickles down the partitions of the holds and on the ammunition. This very grave disadvantage can only be avoided by cooling the air entering the ammunition hold, so that it is not at a higher temperature than that of the contents or of the walls of the hold.

The only rational means to cool ammunition holds is to ventilate them with air suitably cooled, and this deduction has received the sanction of actual practice, as it is the only method of cooling which has prevailed so far. It has been applied to fourteen French battleships, which carry together forty-three ammunition-hold cooling plants, and to eight Russian battleships, which are fitted with thirty-seven similar plants. It will be fitted also to the armored cruisers *Waldeck-Rousseau* and *Michelet*, now in course of construction. The ammunition holds of these two ships are to be provided with æro-refrigerators, in which a circulation of artificially-cooled liquid is maintained constant; the dynamo rooms are to be cooled by a similar installation, but with sea-water circulation.

Until recently the maximum temperature which was thought advisable for ammunition holds was 35 degrees centigrade (95 degrees Fahrenheit); now, however, this limit has been brought down to 30 degrees centigrade (86 degrees Fahrenheit). With the former limit direct cooling by sea-water circulation sufficed, as the temperature of the water

taken from a certain depth remained perceptibly lower than 35 degrees centigrade in all parts of the world. The latter limit, however, can only be reached by having recourse to artificial refrigeration.

The refrigerators for cooling the air consist of metallic surfaces, on one side of which the cooling liquid circulates, the air to be cooled circulating on the opposite side. A pump ensures the circulation of the liquid, and a fan that of the air. The complete apparatus are built by Mr. F. Fouché, 38 Rue des Ecluses St. Martin, Paris.

The extent to which refrigeration has to be carried depends upon the amount of heat which enters the hold; it is necessary, in the first place, to reduce to the lowest possible minimum this amount of heat. The afflux of heat in the hold is caused by radiation from the warm sides and by conductivity from the metallic pieces they contain, and which are connected to the heated sections of the ship. The methods of obtaining a thermic insulation of the hold are the following: Radiation from the warm sides can be reduced by an inside lining made of substances the conductive and emissive properties of which are low, such as cork and asbestos. Another solution is to build a double wall, to obtain an air lining, in which case it is advantageous to ensure in the double wall a circulation of air at the lowest possible temperature. The circulation of a cold liquid inside the double wall will give most satisfactory results, provided the liquid be brought sufficiently in contact with all the metallic pieces that are liable to carry heat into the hold by conductivity.

It is easy in principle to ensure the thermic insulation of the hold, but in actual practice the application of the various methods is surrounded with great difficulties, owing to the arrangement of the holds, the small space available, and the necessity of preventing every cause of damage to the ammunition. The use of simple insulating inside coverings of a thickness proportional to the heating of the walls and the available space is evidently the most easy solution of the problem. As this method has hitherto proved of sufficient

efficiency, it has alone been developed in actual practice. The use of double walls with air circulation has, however, been successfully combined with the insulation of the sides.

The use of a cold-liquid screen is surrounded by the following difficulties: In order to be efficient, there should be no break or interval in the current. It should reach the whole of the metallic pieces which are liable to cause heat in the hold by conductivity. All risks of inundating the hold in case of leakages have to be removed. The cold screen is constituted by a double wall, but it is absolutely impossible to give the space sufficient dimensions to allow a man to enter it in order to paint the plates and keep them in a good state of repair, this proscribing the use of all liquids, such as sea-water, which may corrode the plates. The solution consists in causing to circulate inside the double wall a liquid, such as fresh water charged with oil, or milk of lime, with which there is no risk of corrosion. The total quantity of liquid thus circulating has to be as low as possible, so as to prevent in any case the risk of inundation of the hold. If a leak did occur, a small quantity of liquid only would escape without risk of damaging the ammunition. Simple means suffice to show when an accident of this kind has happened.

The paper was accompanied by drawings showing how a hold could be insulated and the method followed for ventilating it with cooled air. The combination is such that the injected air cannot be hotter than the sides of the hold, which prevents all deposition of moisture. The insulation of the hold is completed by the flowing of the exhausted air round the walls. The liquid and the air for ventilation can be cooled simply by sea-water or by means of a refrigerating machine, according to the temperature required to be maintained in the hold. The amount of heat Q^1 penetrating through the walls is proportional to their surface $\int s$, to the difference of temperature between the two sides $T-t$, and to the time.

The number of heat units going through a given partition per unit of surface and time q^1, for 1 degree difference in

temperature, is determined experimentally. Numerous tests have established the figures corresponding to partitions covered with the usual insulating material. The following formula is obtained:

$$Q^1 = \int s\,(T - t)\,q^1$$

for the unit of time.

Further, the metallic parts, such as partitions, bulbs, standards, projectile supports riveted to a warm wall, cause a quantity of heat to penetrate the hold; this is

$$Q^2 = \int s\,\frac{q^2}{e}\,(T - t),$$

where s is the section of the metallic part considered; e the distance between the point where the heat maintained at a temperature T penetrates, and the point whence this heat is emitted inside the hold, the temperature of which is t; q^2 being the number of heat units passing per unit of time and per unit of section of the piece, considered between two points, with the unit of length between them, and maintained at a temperature difference of 1 degree.

Neglecting the amount of heat brought in by the *personnel*, caused by the handling of the projectiles, also the entrance of warm air, the total quantity of heat penetrating the hold will be

$$Q = Q^1 + Q^2.$$

This amount of heat has to be totally carried away by ventilation in order that the hold may be maintained at the required temperature. It is necessary, therefore, to send into the hold a volume of air V capable of absorbing the quantity of heat Q, by being heated from its entrance temperature τ to an outlet temperature θ. The calorific capacity of air being 0.3 heat unit per cubic meter, the following will be the formula:

$$V = \frac{Q}{0.3\,(\theta - \tau)}.$$

It is necessary to ascertain that the volume of air thus determined meets the ventilating conditions of the hold according to the nature of ammunition it contains.

In order to bring the volume of air V from the initial temperature θ measured at the moment it enters the refrigerating machine to the temperature θ measured at the outlet, the amount of heat given up by the air proper, by the water vapor it will contain after cooling, and that given up by the vapor which condenses, have to be deducted.

The specific heat of dry air being 0.3 heat unit (calorie) per cubic meter,

That of vapor being 0.48 heat unit (calorie) per kilogramme,

The latent heat of water evaporation being 606.5 heat units (calories) per kilogramme,

p the weight of water vapor contained in 1 cubic meter of air drawn, and

p^1 that remaining in 1 cubic meter of the air after cooling,

The formula for the heat to be absorbed will be

$$Q^1 = V[(\theta - \theta)(0.3 + 0.48p^1) + 606.7(p - p^1)].$$

In designing the refrigerating machine it is necessary to add to this amount of heat Q^1 that resulting from the heating of the apparatus themselves, and of the pipes in which cold air and liquid circulate.

The dimensions of the apparatus and of its auxiliary parts—the circulation pump and the fan—are deduced from the volume of air V flowing per unit of time, and from that of the cold liquid necessary to carry away the quantity of heat Q^1 increased as aforesaid.

It should be remarked that the power of transmission of heat of an æro-refrigerator from air to water is, so to speak, without limits; it depends solely upon the output. It increases with the speed, so that the dimensions of the apparatus, suitably proportioned, are solely dependent upon the required pressure of the air and cooling liquid.

Observations made on a number of installations show that the machines supply air easily at a temperature which does not exceed by much more than about 1 degree centigrade

that of the cooling liquid, whatever be the temperature of the air drawn into the machine. The water vapor contained in the air is completely condensed in the æro-refrigerator in a proportion which corresponds with the fall in temperature. The hold thus remains perfectly dry.

Tests made on board the *Sully*, in the Far East, have shown that with temperatures of 31 degrees centigrade (87.8 degrees Fahrenheit) for sea water, and 36 degrees centigrade (96.8 degrees Fahrenheit) for the air, the apparatus maintained the hold at a temperature of 32 to 34 degrees centigrade (89.6 to 93.2 degrees Fahrenheit).

The results of the other tests, and illustrations of the apparatus, were given in the paper.—"Engineering."

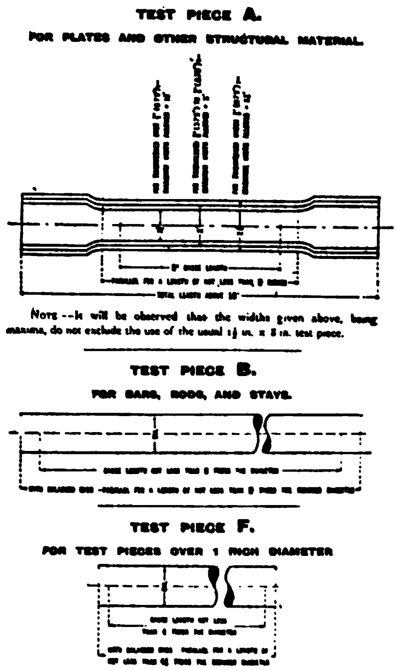
BRITISH STANDARD SPECIFICATION FOR INGOT-STEEL FORGINGS FOR MARINE PURPOSES.

Ingot steel for forgings for marine purposes may be made by the open-hearth process, acid or basic, as may be specified, and as approved by the inspecting body.

The forgings must be made from sound ingots, and for all important forgings (such as crank and propeller shafts, connecting rods, piston rods, crosshead pins, etc.) the forgings must be gradually and uniformly forged. Not more than the lower two-thirds of the ingot is to be utilized for purposes of forging. The sectional area of the body of the forging (as forged) shall not exceed one-fifth of the sectional area of the original ingot, and no part of the forging (as forged) shall have more than two-thirds of the sectional area of the original ingot.

The tensile strength of ingot-steel forgings for marine purposes, ascertained from standard test pieces, must be between the extreme limits of 28 and 40 tons per square inch. In all cases a margin of four tons per square inch shall be allowed between the specified maximum and minimum tensile breaking strengths. The elongation, measured on a standard test piece, must be not less than 29 per cent. for 28-ton steel and

17 per cent. for 40-ton steel, and in no case must the sum of the tensile breaking strength and corresponding elongation be less than 57.



STANDARD TEST PIECES.

Cold-bend tests are to be made upon test pieces having a rectangular section of 1 inch wide by $\frac{3}{4}$ inch thick. The test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$ inch. The test pieces shall be bent over the thinner section. Bend tests may be made by pressure or by blows.

The test pieces must withstand, without fracture, being bent through an angle of 180 degrees, the internal radius of the bend being not greater than that specified below.

Maximum specified tensile strength of forging.	Internal radius of test piece after bending, Inch.
32 tons per square inch, ...	$\frac{1}{4}$
Above 32 and up to 36 tons per square inch,	$\frac{3}{8}$
36 and up to 40 tons per square inch,	$\frac{5}{8}$

BRITISH STANDARD STEEL BARS, ROUND AND SQUARE.

Nominal size of bar. (Diameter or width across flats.)	Hot rolled. Margin of manufacture.	Bright rolled and drawn. Margin of manufacture.
Inches.	Inch.	Inch.
$\frac{1}{4}$ (.25),	+ .010	— .002
$\frac{5}{16}$ (.3125,)	+ .010	— .002
$\frac{3}{8}$ (.375),	+ .010	— .002
$\frac{7}{16}$ (.4375),	+ .010	— .002
$\frac{1}{2}$ (.5),	+ .010	— .002
$\frac{9}{16}$ (.5625),	+ .010	— .002
$\frac{5}{8}$ (.625),	+ .010	— .002
$\frac{3}{4}$ (.75),	+ .010	— .002
$\frac{7}{8}$ (.875),	+ .010	— .003
1,	+ .015	— .003
$1\frac{1}{8}$ (1.125),	+ .015	— .003
$1\frac{1}{4}$ (1.25),	+ .015	— .003
$1\frac{3}{8}$ (1.375),	+ .020	— .003
$1\frac{1}{2}$ (1.5),	+ .020	— .004
$1\frac{5}{8}$ (1.625),	+ .020	— .004
$1\frac{3}{4}$ (1.75),	+ .020	— .004
$1\frac{7}{8}$ (1.875),	+ .025	— .004
2,	+ .025	— .004
2 to 4 inches, black square,	+ .030	...
2 to 3 inches, bright round,	...	— .005
2 to 6 inches, black round,	+ .025	...

GERMAN NAVAL POLICY AND THE NORTH SEA CANAL.

A scheme necessary for the realization of the ambitious naval scheme of the German Government is the widening of the North Sea Baltic canal, which will involve a very heavy expenditure. The canal at present is of such dimensions as to limit the size, and particularly the width and draught, of battleships, and the advent of the *Dreadnought* made it imperative on the part of Germany, if her warships were to equal the new types, to practically reconstruct the canal. To

this extent the Government will be handicapped in their effort to provide the funds for the building of ships. The extent of their expenditure on works will be appreciated by a description of the canal scheme now before Parliament.

The section of the new canal, prepared in connection with the Government Bill before the German Parliament, shows a depth of 11 meters (about 36 feet) below average canal water level, and at this depth the breadth is to be 44 meters (about 144 feet); whilst the present depth is 2 meters less, and the breadth only half the proposed breadth. The breadth at the surface of the water will be about 340 feet, against 223 feet as at present, and the water-carrying section is increased from 413 square meters to 825 square meters. This profile should answer all requirements, as far as these can at present be foreseen.

These dimensions will allow of the bridges at Grünenthal and Levensau remaining, in spite of their comparatively small foundation depth. Should, at any time, a further enlargement of the canal section be required, the depth can, without difficulty, and without a very material increase in cost, be increased to 13.5 or 14 meters. Under such circumstances the two high-level bridges already referred to would, however, have to be rebuilt. In order to avoid this contingency for the new high-level bridges the foundations of the latter will be carried to a greater depth. So as not to increase too materially the cost of the extension the present line of canal will, on the whole, be adhered to.

In two places, the Obereidersee section and that between Holtenau and Levensau, a new line will be cut, so as to avoid some undesirable curves. These alterations in the direction are looked upon as indispensable. The former section (the Obereidersee) has proved unsatisfactory ever since the construction of the canal. The high banks at the curves interfere with navigation by blocking the view and the sight of approaching steamers, so that it is difficult to ascertain their speed. The section between the Levensau Bridge and the Holtenau Lock is equally awkward, inasmuch as the sharp-

ness of the curve places obstacles in the way of the passage of the large steamers, and proper regard to safe transit makes the doing away of these curves essential. The smallest permissible curve should be 1,800 meters radius, whilst the one on the Levensau-Holtenau section is only 1,200 meters. The canal line must be straight on either side of the Levensau Bridge, as the section of the bridge for large steamers is only 38 meters broad. It is also essential that vessels coming from the west should have an open view of the Holtenau inner harbor and locks. Similar considerations make it also desirable that the canal should be made distinctly broader at other sharp curves.

For the purpose of increasing the efficiency of the canal the distance between the passing places will in the future be only a little more than six miles (10 kilometers). The experience gathered from the canal traffic in the past shows that this is the shortest distance there should be, and even that is considerably more than at several other canals; at the Suez canal the distance is only about three miles. Four of the passing places will be so constructed as to allow of turning, an arrangement which will allow a fleet passing through the canal to turn in comparatively short time. For the ordinary passing places the bottom breadth is about 445 feet, and the breadth at surface of water 635 feet. At those intended for turning places the length is 3,670 feet, the breadth at bottom 550 feet, and at surface of water 730 feet, with a turning circle of 1,000 feet in diameter at the bottom.

The locks at Brunsbüttel and Holtenau will be 1,100 feet long between the gates, 150 feet broad and 46 feet deep, with ordinary canal water level—that is, Baltic water level. The locks will consequently, even with comparatively low water level, still have a depth of 40 feet. The locks of the new Kaiser Harbor at Bremen are 735 feet long, 93 feet broad and 35 feet deep at ordinary high water. The proposed depth of the canal locks will, as far as can be seen, be sufficient for the largest war and merchant ships. These dimensions are all the more important at the Brunsbüttel lock, on account of its

vicinity to the mouth of the Elbe, where damaged, deep-draught ships can be quickly conveyed to the interior harbor for repairs. In fixing the breadth due consideration has been given to the fact that the breadth is likely to increase in the shipbuilding of the future. Although the lock dimensions, perhaps, may be a little ample for the present time, the margin is looked upon as unavoidable, as a possible later enlargement of the locks would mean complete rebuilding. These large locks may also, it is claimed, prove a boon to the shipyards on the Baltic, as they will allow of the large steamers built there going through to the North Sea.—"Engineering."

BRITISH TARGET PRACTICE.

The cruiser *King Alfred*, flagship of Vice-Admiral Sir Arthur Moore, Commander-in-Chief of the China Squadron, has made the following scores while at gunnery practice at Wei-hai-wei:

With three 6-inch guns in one minute—

Rounds.	Hits.	Bulls.
11	11	11
14	13	8
13	13	9

With her two 9-inch guns in two minutes—

Rounds.	Hits.	Bulls.
10	10	8
9	9	7

Altogether eighteen big guns fired 198 rounds making 188 hits and 113 bulls. This remarkable record places the *King Alfred* far ahead of every ship in the British Fleet.

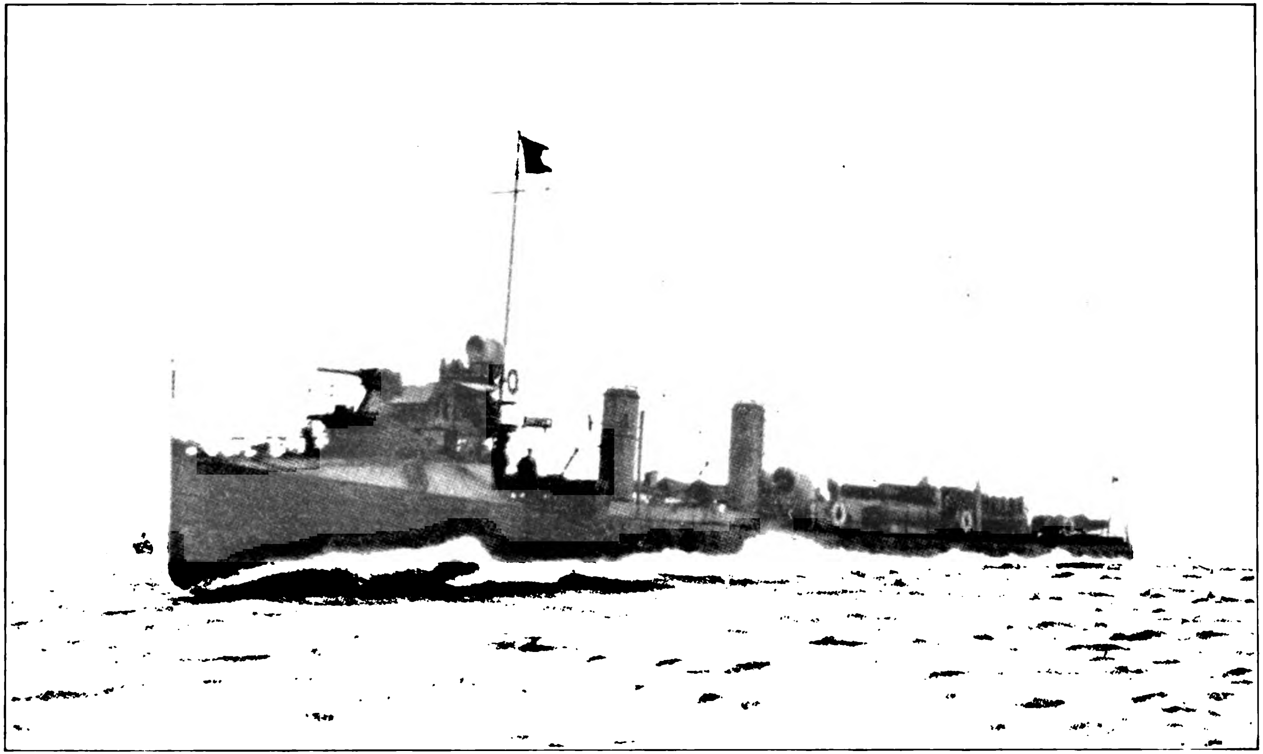


Fig. 1.—The Torpedo-Boat Steaming at 27 Knots.

OIL-FUEL TURBINE-DRIVEN TORPEDO BOATS.

There are given here several illustrations of the 26-knot torpedo boats built by Messrs. J. Samuel White & Co., Ltd., of East Cowes, Isle of Wight, for the British Navy, and there is reproduced a plan of the machinery room and another view of the high-pressure turbine. These represent a new series of boats, of which twenty-four have been laid down since the first were ordered under the naval programme of 1905-6. Up to that time the torpedo boats had ranged in speed up to 25 knots, the largest being of 205 tons displacement, with a length of 165 feet, 17 feet 6 inches beam, and 5 feet 6 inches draught; the horsepower of the reciprocating machinery being 3,000. Those vessels were completed three or four years ago. Later it was considered that a superior type of boat was necessary for even the tactical work undertaken by purely torpedo craft, in contradistinction to the over-sea offensive work of destroyers. The Board of Admiralty therefore decided to introduce a new type, which were at first classed as "coastal torpedo-boat destroyers," but this has since been changed to torpedo boats. In view of the great suitability of the Parsons turbine, as ascertained in exhaustive trials with successive destroyers—notably the *Eden*—this type of propulsive machinery was adopted. At the same time the Committee appointed over to investigate the advantages of oil fuel had solved the difficulties of the system, not only as regards the best means of vaporizing the fuel, but also of completely consuming it, and the new boats were therefore designed with boilers to use only liquid fuel. As we have said, twelve boats were ordered, the design being left to the three firms with whom the contracts were placed.

Messrs. J. Samuel White & Co., Limited, of East Cowes, secured the order for five, which was not surprising to those cognizant of the work done by the Isle of Wight firm, not only in connection with torpedo craft, but with high-speed light-draught vessels. The steady development of the torpedo boat, from the advent of the *Lightning*, about thirty years ago, had forced upon the minds of tacticians the neces-

sity for a vessel of the same type and of greater power to act as catcher or destroyer, and Messrs. J. Samuel White & Co., of Cowes, were the first to produce such a catcher, one vessel —the *Swift*—passing through successful trials in 1885, while another, two years later, of the same type but of larger dimensions, proved equally successful, and was purchased by the then firm of Sir W. G. Armstrong, Mitchell & Co., and subsequently added to the Chinese Navy. Since then the firm have kept pace with the requirements of the naval strategist, and proof of their success is found in the fact that out of thirty-two of the modern torpedo craft with oil fuel and Parsons turbines, ranging from 26 to 36 knots speed, now under order, eleven are being built by the Isle of Wight firm.

When the order for the vessel illustrated was placed, the firm were in an advantageous position for carrying out the undertaking. They held a license from Messrs. Parsons for turbine construction, but although no such engines had been made, new machine tools were rapidly installed for the purpose. This, however, was a matter of comparative simplicity. What was of more consequence was the necessity of securing the minimum scantlings for the hulls with absolute oil-tightness in those parts in which the fuel was to be stored. A complete series of experiments were entered upon, culminating in the construction of a short length of vessel, of the full section, in order to practically test the system of riveting proposed. This enabled the constructional details to be settled, and it is interesting to note that other firms followed the same lines, and a paper on the subject was read at the Institution of Naval Architects last year dealing with experiments of the same description as those previously carried out at Messrs. White's works (see "Engineering," vol. lxxxi, pages 511 and 802). The details of scantling thus having been determined, rapid progress was made with the construction of the first five vessels. All the boats were delivered before the contract date, the first anticipating the time by a month, and the fifth, which was handed over on the 8th of May last, by nearly three months.

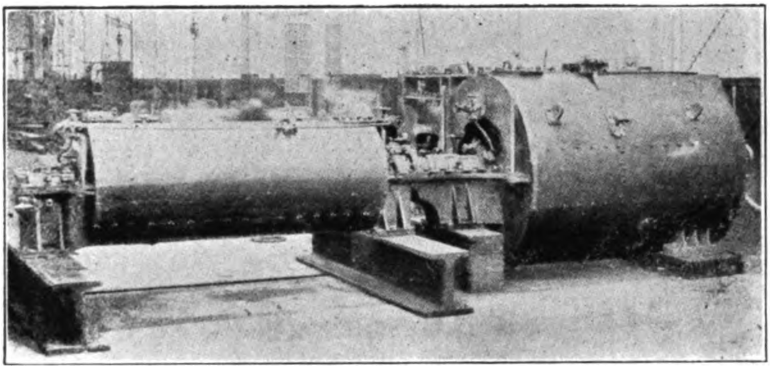


Fig. 2.—The Cruising Low Pressure, and Astern Turbines.

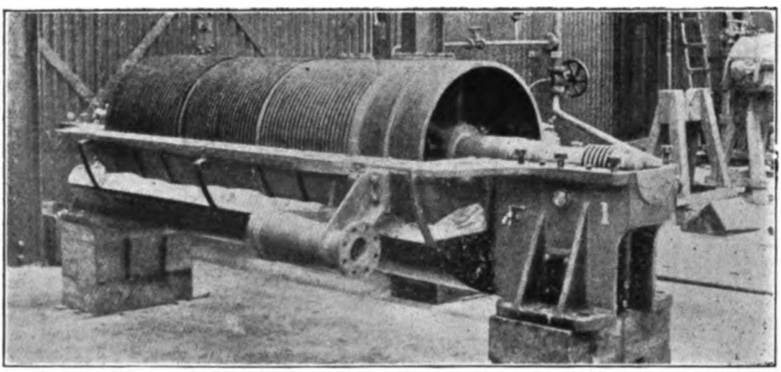


Fig. 3.—The Cruising Turbine with Top Cover Off.

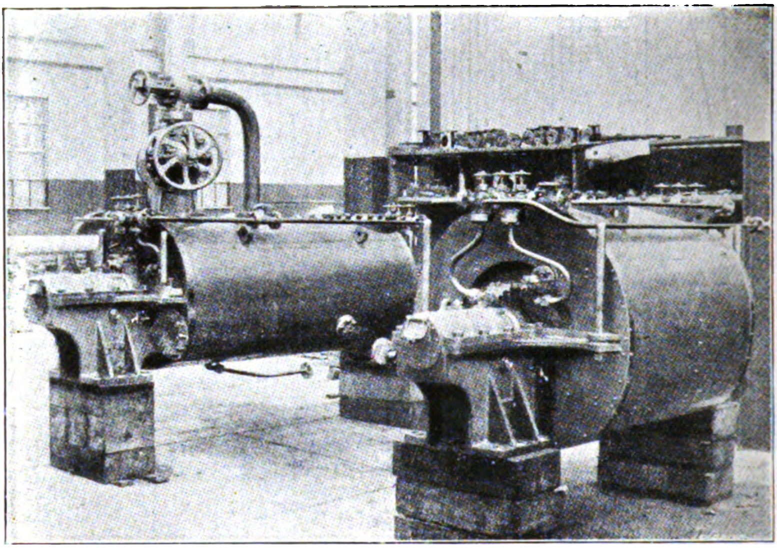


Fig. 4.—The High-Pressure and Intermediate-Pressure Ahead Turbines.

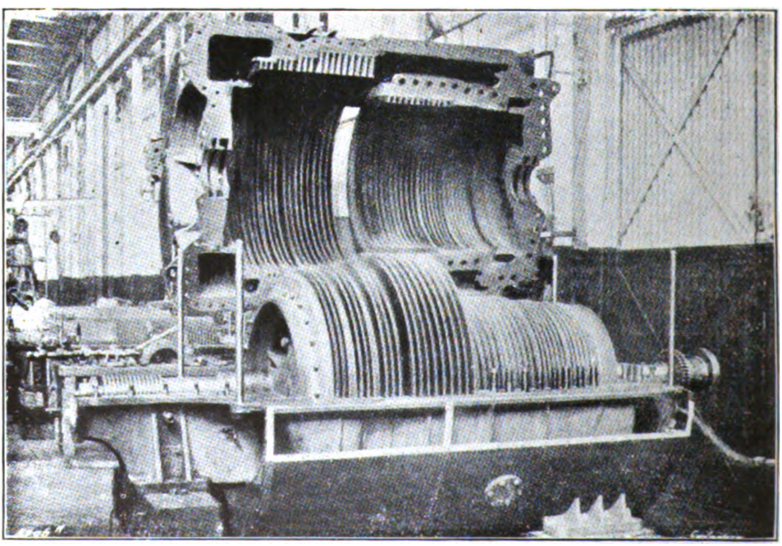


Fig. 5.—The Low-Pressure Ahead and the Astern Turbine (Combined).

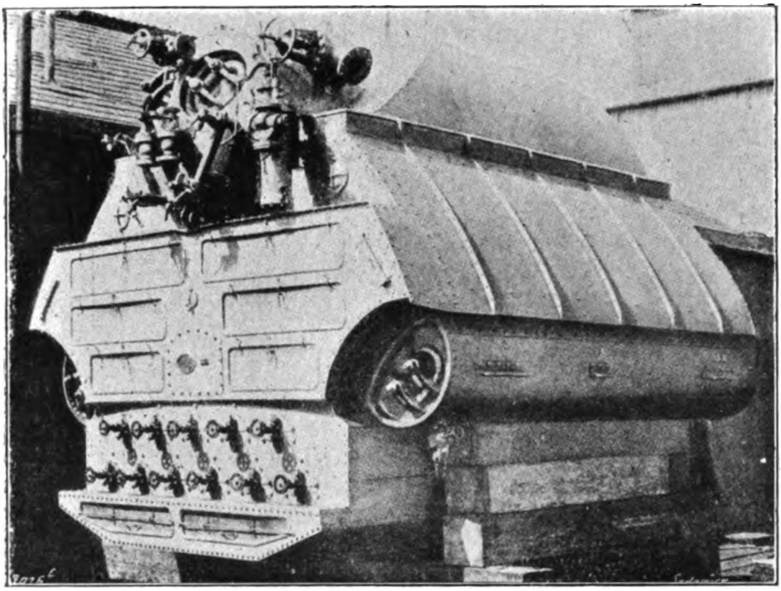


Fig. 6.—The White-Forster Water-Tube Boiler.

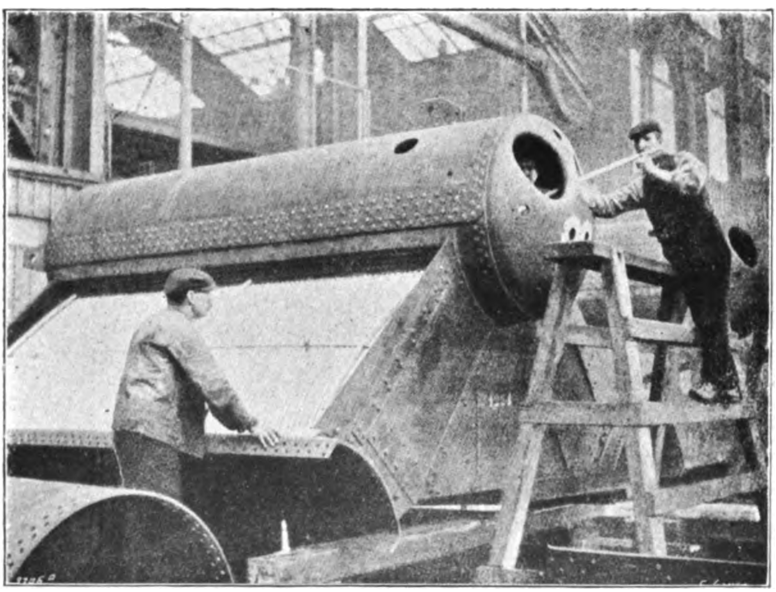


Fig. 7.—Method of Tubing White-Forster Boilers.

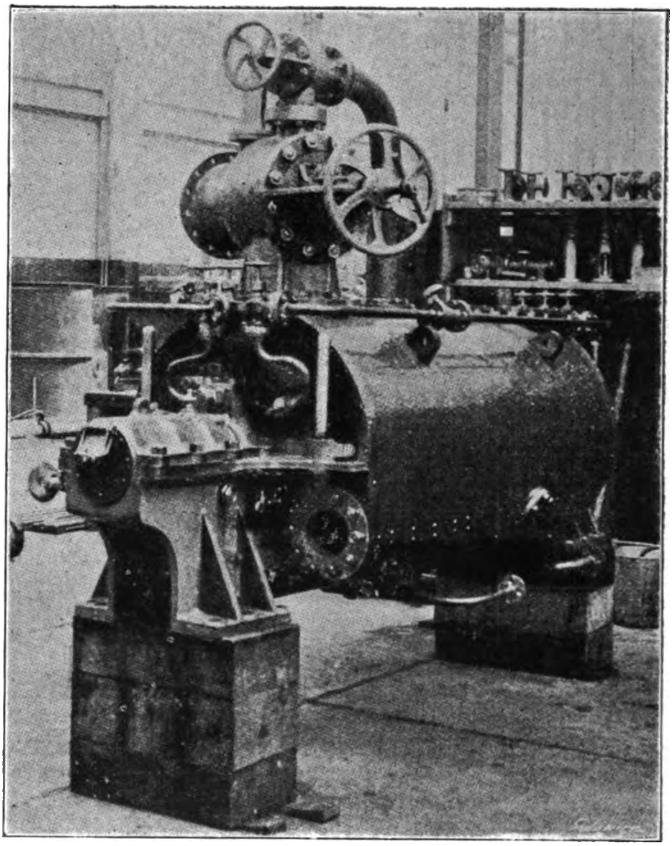


Fig. 9.—HIGH-PRESSURE AHEAD TURBINE.

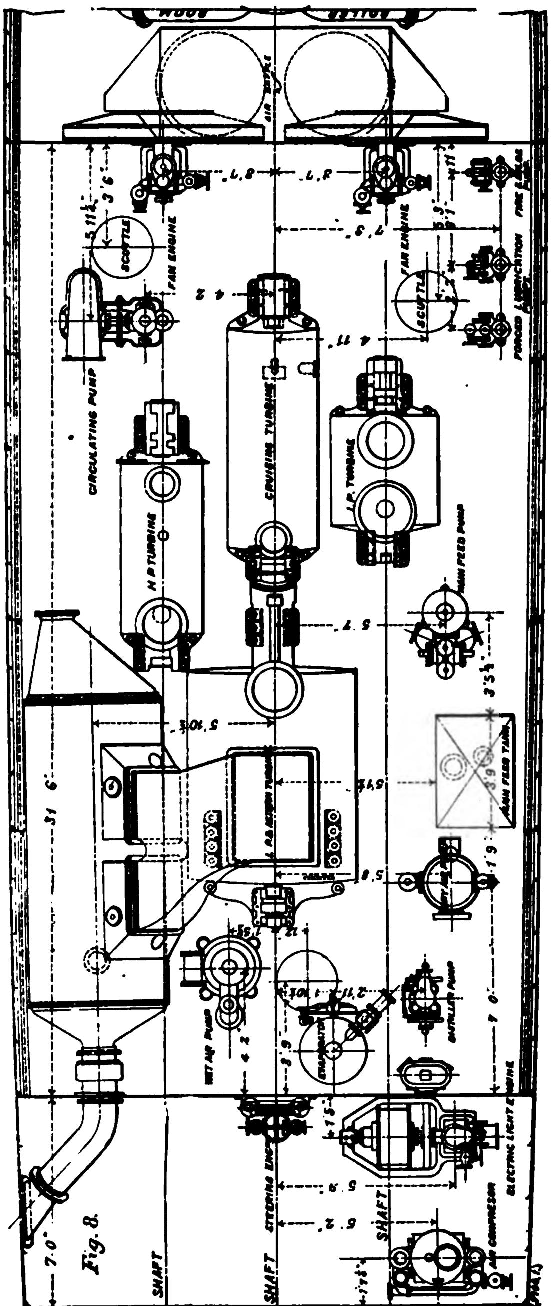

CIRCULATING PUMP
FAN ENGINE
SCUTTLE
H.P. TURBINE
CRUISING TURBINE
L.P. TURBINE
MAIN FEED PUMP
MAIN FEED TANK
WET AIR PUMP
DISTILLER PUMP
STEERING ENGINE
SHAFT
AIR COMPRESSOR
ELECTRIC LIGHT ENGINE

Fig. 8.

The first five vessels built by Messrs. White were 175 feet long, 17 feet 6 inches beam and 10 feet 9 inches in depth, the displacement at 5 feet 8 inches draught being 245 tons. The vessels, as shown in Fig. 1, are fitted with a high whale-backed forecastle, with a bridge carrying a 12-pounder gun, while the three torpedo-launching tubes are mounted at intervals on the middle line; there is another 12-pounder gun aft. The feature of the ships, however, is the steam-generating plant and turbine machinery.

The boilers adopted by the firm are of the well-known White-Forster type. Two illustrations showing the distinctive characteristics of this boiler are given (Figs. 6 and 7), one of which shows the work of putting in the tubes, which are of 1-inch and 1⅛-inch diameter, while the other shows the completed boiler with all its fittings and with the liquid-fuel nozzles. The details of the oil-fuel installation—the result of costly experiments by the Admiralty—are very properly kept secret. The relation of grate surface to heating surface in the White-Forster boilers in these vessels is 1 to 60 square feet, and the practice in liquid-fuel installations is to provide 2 square feet of heating surface for the equivalent of 1 indicated horsepower. The boilers have proved very satisfactory in these vessels, to which fact testimony is borne by the boiler being bracketed as an alternative type with the Yarrow boiler in the specification for the new high-speed cruiser *Boadicea*, which is being built at Pembroke as a mother boat for torpedo craft. There are two boilers in each torpedo boat, and these are accommodated in one stokehold, the fans for which are, as will be seen in Fig. 8, driven by engines in the engine room under the immediate observation of the engineer. The shaft passes through a stuffing box in the bulkhead.

The general plan shows the engine arrangement, and on the other plates there are several views of the turbine in course of construction. On the center shaft there is a cruising turbine and a low-pressure ahead and an astern turbine, the latter two within one casing. On the port wing shaft there is a high-

pressure ahead turbine, and on the starboard side an intermediate-pressure ahead turbine. All four ahead turbines are in use for speeds up to 16 knots, steam from the boiler being passed to the cruising turbine, thence to the high-pressure turbine on the port shaft, to the intermediate-pressure turbine on the starboard shaft, to the low-pressure turbine on the center shaft, and, finally, to the condenser, which is placed on the port side of the ship. This arrangement of four compound turbines gives a great range of expansion, and has resulted in satisfactory economy even at low powers. For speeds above 16 knots the steam is shut off from the cruising turbine—which then works *in vacuo*—and passes direct to the high-pressure turbine, thence to the intermediate, the low-pressure, and finally to the condenser. The turbines, which were constructed by Messrs. J. Samuel White & Co., are bladed on the Parsons system, and the details correspond with those generally adopted in torpedo craft. Fig. 2 shows the cruising turbine, with the low-pressure ahead and astern turbines to the rear. Fig. 3 illustrates the rotor of the cruising turbine, from which it will be seen that there are three stages. Fig. 4 shows the high-pressure and intermediate turbines, which are considerably shorter than the cruising turbine. On this view there is seen the valve for the passage of steam from the boiler direct to the high-pressure turbine. Fig. 5 is specially interesting, as it shows the rotor of the low-pressure and astern turbines in place, with the upper part of the casing raised, and the thrust shaft and vertical supports for lifting the upper part of the casing. Fig. 9 illustrates the high-pressure turbine complete, with its steam valves.

The contract called for the maintenance of a speed of 26 knots during an eight-hours' run, with 21 tons of oil fuel on board, and all five vessels easily maintained this condition. The mean results of the five boats were as follows:

Speed on measured mile (Stokes Bay), knots	27.107
Mean speed for eight hours, knots	26.36
Consumption of oil for eight hours, tons	20.758
Distance run per ton of fuel at a speed of 12.147 knots, nautical miles.	37.90

It should be noted that as the depth of water at Stokes Bay is not quite adequate for such high speeds the vessel will, in deep water, exceed the 27.107 knots attained on the measured mile. The consumption of oil averaged just under 2.6 tons per hour for 26.36 knots. On the earlier torpedo boats of 25 knots speed, of which Messrs. J. Samuel White & Co. built several, the consumption was also 2.6 tons per hour, but the speed was lower (1 mile per hour), and the displacement was 30 tons less than the new oil-fuel turbine-driven boats, so that the result must be pronounced very satisfactory. It will be noted also that at 12 knots the vessels steamed 37.99 nautical miles for each ton of fuel consumed, whereas the guarantee under the contract was 32 nautical miles, so that here, as in the matter of speed, there was a considerable margin over the stipulated condition.

Messrs. J. S. White & Co. are building four more vessels of this type; the dimensions have been slightly increased—the length by 7 feet, the beam by 6 inches and the displacement by 10 tons—so that the new vessels are 182 feet long, 18 feet beam, and at 5 feet 10 inches draught they will displace 256 tons. The estimated power to give them 26 knots is 4,000 horsepower. Two 33-knot destroyers are also in course of construction. These are 272 feet in length, have a beam of 26 feet, and at 8 feet 8 inches draught the displacement is expected to be about 900 tons. It is anticipated that the 33 knots will be realized with a power equivalent to 15,500 indicated horsepower. Five other vessels of this class are being built, in addition to a 36-knot destroyer by other firms, and the performances of these various vessels on trial will be looked forward to with considerable interest.—"Engineering."

RECIPROCATING ENGINES FOR OCEAN-GOING STEAMERS.

Paper read at the Engineering Conference of the Institution of Civil Engineers, Section III, Machinery, June 19, 1907, by HENRY DAVEY, M. Inst. C. E.

The propelling power of the ship begins with the fuel and ends with the screw. The real fuel economy is the fuel required to drive the ship at a given speed. If the turbine requires a screw which is necessarily less efficient than that of the piston engine, it is the fault of the turbine system. Leaving out the boiler efficiency, the propeller of the ship is the engine and screw combined. In this connection, what is wanted for the purpose of ascertaining detail losses is not only the actual mechanical efficiency of the turbine, which Mr. Denny is trying to obtain by measuring the torsion of the shaft, but the actual thrust exerted by the screw in propelling the ship. I think an indicator for that purpose might be devised, but it should be an indicator for use with the ship in motion under normal conditions.

The subject of ordinary reciprocating, or, to use a shorter word, piston, engines can better be discussed by comparison with rival systems, such as turbine and gas or oil engines, although the two latter are not at present used for large ocean-going steamers.

On the question of fuel economy alone I have not been able to obtain any reliable information which would enable me to compare the piston engine with the turbine; but it does appear that the turbine has not beaten the piston engine in over-all efficiency—that is, taking engine and screw together.

Economy of fuel must always be one of the most important factors in steamship working, and it may be that the competition of the turbine will lead to improvement in the fuel economy of the piston engine, and possibly in an improved system of working. The turbine will doubtless be improved also.

There is little hope of improved fuel economy from increased boiler pressure alone. The theoretical possibilities of

saturated steam at 100 pounds, 150 pounds and 200 pounds, and at 27.5 inches vacuum, are $8\frac{3}{4}$ pounds, 8 pounds and $7\frac{1}{2}$ pounds of steam per indicated horsepower respectively. The best land engines give an efficiency ratio of about 70 per cent., corresponding with 12.5 pounds, 11.4 pounds, and 10.6 pounds respectively. That is a much better economy than is usually obtained with high-class marine engines.

High-class land engines, owing to the use of the Corliss and drop types of steam valves, have much less clearance space; they have also a much better steam distribution, and often have reheaters between the cylinders. I do not see why Corliss or drop valves should not be used in marine engines. They might be placed in the covers of the cylinders, and the clearance spaces would in that way be greatly reduced. Reheating between the cylinders might also be adopted on a plan to be named further on.

The introduction of the turbine has given rise to a new arrangement of engines, an arrangement which appears to me to be advantageous for piston engines also—namely, the arrangement of three screw shafts with a turbine on each shaft, the center turbine, or high-pressure one, exhausting into, and supplying the steam for, the two low-pressure turbines working the side, or wing, screws. This arrangement of engines and distribution of steam appears to me to be suitable also for piston engines. Each piston engine would have a separate reversing gear, and the gears would be actuated separately or together as desired. Another arrangement would be that of having a high-pressure piston engine for the center screw, exhausting into low-pressure turbines working the wing screws. It would occupy too much space to go into details, but the scheme is worth consideration, as it would lend itself to a system of superheating the steam between the high and low-pressure engines. In many land engines steam is superheated between the cylinders by means of reheating coils supplied with boiler-pressure steam, but the amount of heat thus added to the steam is not sufficient to effect much saving. What is wanted is an effective superheater for the

purpose. It does not appear practicable to supply the heat from the boiler, but a separately-fired superheater, fired with gas from a gas producer, does not appear to present the same difficulties. An ordinary gas producer without gas-cleaning plant would answer the purpose. If a practicable system of what I will call stage superheating could be worked out, greatly increased economy would result from its application to both piston and turbine engines.

The piston engine appears to be able to compete favorably at present with the turbine in every respect, except that with the turbine there is not the thud of the piston engine, there is probably much less vibration, and there appears to be a considerable reduction in the engine-room staff. Also, the cost of upkeep may be less; but only time and extended experience can satisfactorily determine these questions. I do not know that the vibration due to screws alone would be more with the piston engine and three screws than it is with the turbine and three screws. Probably, as the cost of fuel is so important an item in the total cost, the race between the turbine and the piston engine will be won on the coal bill.

Use of Gas Engines on Board Ship.—I am of opinion that there is little hope of the marine gas engine proving more economical in fuel than the piston or turbine engine. Apart from the question of fuel, the mechanical difficulties connected with the use of large gas engines for such a purpose are very formidable. If bituminous coal is used, a very extensive gas-cleaning plant will be required, a plant quite unsuitable for use in a ship. If the gas is not thoroughly cleansed, tar gets into the engine and gives rise to troubles which could not be put up with during a voyage. The consumption of coal would probably be about $1\frac{3}{4}$ pounds per brake horsepower, a result which might be easily beaten by the present or improved piston engines or turbines.

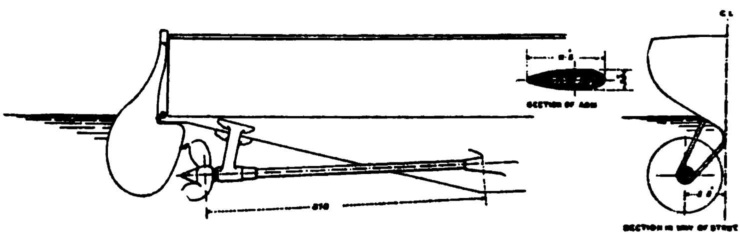
A better use of gas on board ship would be for the purpose of stage steam superheating for piston or turbine engines. A gas-cleaning plant would not be required in that application. Assuming the piston or turbine engine to be made

capable of working with an economy of 10 pounds of steam per indicated horsepower hour, the result would be much better than that likely to be obtained from the gas engine, even if the engine could be practically applied and made satisfactory in other respects.—"Engineering."

PROPELLER STRUTS.

Read before the Institution of Naval Architects, March 21, 1907, by GEORGE SIMPSON.

The subject of suitable area for propeller-strut arms is one which, so far as the writer is aware, has not hitherto been dealt with in the transactions of this or kindred institutions, and has, for some reason or other not explained, been omitted from the rules of the classification societies to whose requirements so great a proportion of our vessels are built. With



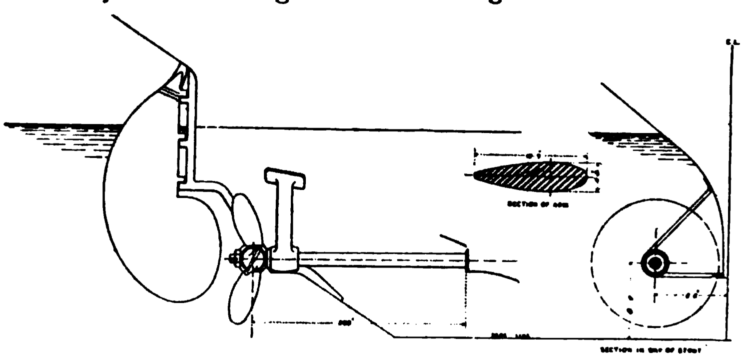
STRUTS FOR THE PROPELLER SHAFTS ON A TORPEDO BOAT.

the introduction of turbines for marine propulsion, and the consequent increase in the number of shafts, coupled with the fact that these structural fittings are of at least equal importance to rudder stocks, whose diameters have always been the subject of specific requirements, the subject of proportional sizes for these struts becomes worthy of interest. The object of the paper is to induce discussion on a semi-empirical formula, whose function is the determination of suitable dimensions for strut arms by a definite method, with a view to insuring proportional relationship between the area of arm, the horsepower transmitted and the overhang of struts.

The present almost general practice of proportioning these directly on the horsepower and previous experience, or in

many cases individual opinion, has resulted in widely divergent results for similar work. This is probably more particularly noticeable in the size of strut arms for the larger classes of naval vessels, where in many cases we find about twice the sectional area which would be assigned for an ocean liner of similar power, revolutions and overhang, although structurally the naval vessels are of lighter construction. The result is that many tons of valuable material and weight are lost to the builder and owner respectively.

It would seem almost impossible to determine the stresses exerted on these strut arms, although the impression at first sight tends to the view that the pressure exerted on the boss journal by the overhang of arm would give a moment suffi-



STRUTS FOR THE PROPELLER SHAFTS ON A TWIN-SCREW FREIGHT STEAMER.

ciently accurate to calculate an equivalent resisting moment of section with a suitable fiber stress. This view of the case, considering the bracket as a cantilever, would result in giving the greatest section modulus to the arms where they adjoin the shell palm, and tapering from thence to the junction with the hub. This was not an unusual design of struts when twin-screw vessels first came into general use. The few cases, however, in which propeller struts were carried away, showed conclusively that the greatest stresses, apart altogether from accidental causes, were those induced by centrifugal forces, and were alternating ones, causing vibration to the outer part of the arms where they joined the hub. These stresses should

be very slight with a perfectly-balanced propeller, and would be greatly augmented by a badly-balanced wheel or in a vessel of light construction, where there is a tendency for the after part of the vessel to "wiggle." That such a movement does take place to a very appreciable amount came under the immediate notice of the writer on one of the 27-knot destroyer class having a rudder of the usual overhung type, and a stock of exceptionally large diameter and of 36-ton steel, where the distance between the side of rudder and the propeller tips was 1½ inches measured in dock, and after trial trip, on redocking, the propeller tips were found bent over and the rudder plates excoriated with contact. The deflection of the rudder as calculated accounted for only about ⅜ inch of the original 1½-inch clearance, so that the contact of propeller and ship was in a great measure due to this "wiggling."

The formula given comprises the factors P, the horsepower transmitted per shaft; R, the number of revolutions per minute; l, the outboard length of shaft, and a coefficient varying with the revolutions, its function being the adjustment of the area solved to the varying length of strut. This function will be more evident from an example.

As one of the factors in computing P is the number of revolutions per minute, it follows that for a given power there may be a great difference in the revolutions per minute, and consequently in the diameter of propeller, the larger wheel requiring a much longer strut to hold the propeller clear of deadwood, and, on the other hand, a high-turning propeller would take a comparatively short strut. It will be evident from this that, viewed as a cantilever, the shorter strut, having the lesser bending moment, would not require so great a section modulus to produce the equivalent resisting moment. In the example stated we have taken the case of $P = 2,000$, transmitted through a shaft turning at 80 revolutions per minute in a freighter, and a similar power given in a torpedo-boat shaft turning at 400 revolutions per minute. The outboard length l in the first case is taken at 300 inches and at 310 inches in the high-speed vessel, as the latter would have

much finer lines, accounting for a greater distance for the intersection of the center line of shaft with the side of vessel than would be the case in the fuller-lined freighter. We then have—

		For the freighter. *Inches.*	For the torpedo boat. *Inches.*
Area	$= \sqrt[3]{\dfrac{P \cdot R \cdot l}{.0633R}}$	$= 73.0$	24.9
Length of section L	$= \sqrt{5.333 \text{Area}}$	$= 19.7$	11.5
Breadth of section B	$= \dfrac{L}{4}$	$= 4.9$	2.8

If we assume 15 feet as the diameter of the freighter's propeller, and 6 feet 9 inches for the torpedo boat, the diagram produced will show the difference in length of strut. If we were to assume a weight for propeller and outboard length of shaft for each case and apply this weight as a deadload for the purpose of comparing the *relative* intensity of the moments acting on each, we should find a very close agreement between the ratios of these compared with the moduli of section of the arms. Of course, it must not be inferred from this that the writer claims for the formula other than what it is, viz : a rule based on observation and practice, with an attempt, as already stated, to secure a fairer relationship between the scantlings of struts and the work they are designed for.

The vessel referred to earlier in this article afterwards broke both of her propeller struts, with an interval of about three months between the accidents. Of course, it may not be right to attribute this altogether to the wiggling movement of the stern previously mentioned ; but there can be no doubt that it may have been a factor of no small amount in causing the fracture. The original struts of this destroyer were tapered from 7 by $1\frac{1}{2}$ inches to 9 by $1\frac{1}{2}$ inches with a minimum sectional area, where they broke, of 8 square inches. It will be seen on reference to ship No. 28 in the table of dimensions, that by the formula given, these struts should have been 29.8 square inches, and of section 12.6 by 3.1 inches.

The other cases in which there is a great difference between actual struts and the formula are Nos. 4 and 30, and in each of these vessels there is no doubt that the struts are dangerously near the breaking point, the extreme frailty of the arms being quite observable on the ship.

DIMENSIONS OF PROPELLER STRUTS AS FITTED AND BY FORMULA.

No.	Per shaft.				As fitted.		Per formula.	
	P.	*R.*	*l*, inches.	$\sqrt[3]{PRl}$	$L \times B$, inches.	Area, square inches.	$L \times B$, inches.	Area, square inches.
1 C	13,000	126	648	1,020	33 × 10	259	26.2×6.5	128.0
2 C	13,000	123	552	959	27 × 7	148.2	25.6×6.4	123.0
3 C	10,500	138	668	1,037	28 × 8,	172.8	25.0×6.2	120.0
4 S	8,000	200	289	773	20 × 3	45.0	18.1×4.5	61.4
5 C	8,000	154	642	925	24 × 6½	122.2	22.5×5.6	94.8
6 C	2,250	172	372	524	16 × 4	48.0	16.0×4.0	48.0
7 B	9,500	120	319	714	30 × 8	180.0	22.6×5.6	95.5
8 B	8,700	125.5	296	686	24 × 6½	122.2	21.4×5.4	86.4
9 B	8,250	120	357	714	33 × 10	250.0	22.6×5.6	95.5
10 B	7,600	128	321	678	24 × 6½	122.2	21.2×5.3	83.7
11 B	5,000	120	356	597	24 × 9	169.5	20.5×5.1	78.5
12 L	5,350	90	390	573	24 × 6½	122.2	23.2×5.8	100.6
13 CS	3,000	150	350	540	16 × 4½	54.0	17.4×4.3	56.8
14 CS	3,000	192	304	560	18 × 4	56.5	15.7×3.9	46.1
15 PC	4,700	111	360	573	18 × 4½	63.6	20.8×5.2	81.5
16 PC	4,600	105	340	548	18 × 5	70.6	20.9×5.2	82.4
17 PC	4,350	113	340	551	18 × 5	70.6	20.3×5.1	77.0
18 PC	4,000	235	528	792	20 × 6	90.0	19.3×4.8	70.0
19 PC	3,670	106	360	520	18 × 4½	63.6	20.3×5 1	77.5
20 PC	3,000	112	368	498	18 × 4	56.5	19 3×4.8	70.2
21 PC	2,600	130	290	461	18 × 4	56.5	17.3×4.3	56.0
22 PC	1,500	100	169	294	9 × 3½	24.7	15.6×3.9	46.4
23 PC	1,300	125	304	367	15 × 3½	41.2	15.6×3.9	46.4
24 PC	840	156	408	377	12 × 4	37.5	14.3×3.6	38.2
25 F	3,500	125	630	651	18 × 4½	63.6	20.9×5.2	82.25
26 F	2,810	125	337	491	22 × 3½	65.0	18.2×4.5	62.0
27 D	4,200	364	483	904	15½× 2¼	27.0	14.5×3.6	39.3
28 D	2,420	370	380	698	8 × 1½	12.0	12.6×3.1	29.8
29 T*	2,000	375	250	572	7 × 1¾	10.0	11.3×2.8	24.1
30 M†	165	180	110	148	5 × 1½	7.0	8.3×2.0	13.0
31 R	150	400	100	181	5 × 1¼	5.5	6.2×1.5	7.1
32 Y	200	150	180	176	8 × 2	13.0	9.9×2.5	18.5
33 P*	300	600	90	253	5 × 1⅛	4.5	5.9×1.5	6.66
34 P*	250	500	72	208	5 × 1	4.0	5.9×1.5	6.6
35 P*	200	550	80	207	3¾× ¾	2.2	5.6×1.4	5.9
36 P*	93	500	72	149	3½× ⅝	1.8	5.0×1.25	4.7
37 P*	17	450	48	71	2¾× 11/16	1.5	3.6×0.9	2.5
38 P*	10	400	42	55	2 × ⅜	0.6	3.4×0.85	2.2

* Single screw with vertical strut. † Strut was perceptibly too light.

C = cruiser. S = scout. B = battleship. L = Liner. CS = channel steamer. PC = passenger and cargo steamer. F = freighter. D = destroyer. T = torpedo boat. M = miner. R = river launch. Y = yacht. P = pinnace.

In support of the statement that in many cases the area of propeller-strut arms is assumed without due consideration, attention is directed to Nos. 1 and 2 in the table of dimensions, where it is shown that in almost similar cruisers a difference amounting to 75 per cent. is observed between the two vessels, showing that, as the lighter strut has performed its work satisfactorily on service, a weight of about 23,000 pounds of cast steel has been sacrificed by the builder, and its equivalent lost in ammunition space. This is not an isolated case, and it is borne out by many examples.

An interesting point to be noted in comparing the arms of struts required for single-screw vessels, where these are fitted vertically, is the much smaller area which the actual struts have in comparison with the formula results. With horizontal struts the case is reversed—that is, in the majority of cases the formula gives a slightly increased area over the actual brackets, except in the case of battleships and cruisers. This is probably accounted for by the fact that in the vertical position, the strut is theoretically not subjected to stresses as great as when it is arranged horizontally.

It would be extremely difficult to state theoretically what fastening is required through the palms to the shell; but an investigation of a great number of struts showed that the proportion of sectional area of fastenings to area of strut arm ranged from 0.4 to 0.55, and the writer is of the opinion that 0.5 would be a very good proportion for rivet or bolt area, and one that should give an ample margin of strength, as the stresses which these have to bear do not appear to be excessive.

In conclusion, it may be stated that the author has applied this formula for the past eight years to the determination of the propeller struts of a great number of vessels, in which the factors varied to the extremes found in such vessels as pinnaces, destroyers, scouts, yachts, freighters and liners. The table given will show how the results in these cases would have compared with everyday practice, had the scantlings been assumed in the ordinary way, the probability being that

in applying the formula a considerable saving in weight would have been effected.—"International Marine Engineering."

NEW VIEWS OF THE CAUSE OF THE CORROSION OF IRON.

At the recent meeting of the American Society for Testing Materials, Dr. Allerton S. Cushman, of the United States Department of Agriculture, made the first public announcement of the very interesting investigations he has been carrying on for several years on the causes which underlie the corrosion of iron. A number of new points were brought out, among which the most startling are that oxygen plays only a secondary rôle in the rusting of iron and that the best preventatives of rust are to be found among the most effective oxidizing agents known, such as chromic acid and its salts. This view is so contrary to all previous conceptions that it is naturally received with some incredulity when first heard, yet those who are familiar with the investigations and conceptions upon which the new theory of corrosion is based are of the opinion that the evidence which has been brought forward is not only convincing but conclusive. Dr. Charles B. Dudley, whose conservatism in expressing opinions on theoretical subjects is well known, stated at the close of the presentation of Dr. Cushman's paper that he considered this work to be the most important contribution to a metallurgical problem that has been made in the last quarter century. The same opinion was expressed privately by others who are familiar with the work.

The fact that chromic acid and its salts act as inhibitors of rusting has been known for some time, but no explanation of the curious phenomenon has ever been offered heretofore nor has its application to practice ever been suggested. Dr. Cushman has made a special study of this problem, and although it remains to be seen what practical benefits may develop out of these new ideas, it is most gratifying to be able to state that if any patents are granted covering rust inhib-

itors they will be taken out in accordance with the practice of the Department of Agriculture, so that they will be free for all American citizens. The investigation was begun by Dr. Cushman as a part of his study of the corrosion of fence wire, in the interest of the farmer. A farmers' bulletin on this subject was published by the Department of Agriculture about two years ago. It is safe to say that the public benefit which is likely to follow these investigations is on a par with the important economical benefits which will probably be derived from the same author's researches on the decomposition of feldspathic rocks and their use as fertilizers. Both of these investigations, made in the interests of the people, demonstrate the great national benefit which is derived from the scientific research of our federal departments, conducted by specialists of the highest rank. It is to be hoped that Congress will not fail to appreciate the advisability if not the necessity for providing the funds to carry on work of this kind, the expenditure for which is so small in relation to the public benefit.

If a text-book is consulted for an explanation of the rusting of iron it will be found that carbonic acid has heretofore been generally held responsible for the formation of rust. Iron is supposed to be attacked by carbonic acid, with the formation of carbonate, which is then acted on by water and the oxygen of the air to form the red hydroxide known as rust, the carbonic acid being again set free to take up its destructive work. According to this theory, in an atmosphere which did not, like that of this earth, contain about four one-hundredths of one per cent. of carbonic acid, the rusting of iron would be an unknown phenomenon. That this, as well as the peroxide hypothesis which has lately been developed in England, must be relegated to the dump pile of abandoned theories, seems to be conclusively shown by these latest researches.

According to the electro-chemical or electrolytic theory which Dr. Cushman upholds, the first attack on iron is not made by oxygen, even in the presence of water, but by hydrogen in the form of the hydrogen ion. According to the mod-

ern theory of solutions, many substances when dissolved in water are dissociated into ions, which may be regarded as atoms carrying static electrical charges. Water itself, even when pure, contains a certain proportion of hydrogen ions, and the presence of many impurities, especially those which are by nature acid, increases the hydrogen ions and thus the tendency to attack iron and carry on corrosion. The action is entirely electrolytic, being continually accompanied by an exchange of the electro-static relations between the iron and the attacking hydrogen. Such oxidizing agents as the chromate and bichromate of potash inhibit rusting by polarizing the iron to the condition of an oxygen electrode, thus preventing the approach or attack of the hydrogen ion. One of the most extraordinary points brought out is that this polarization effect is to some extent lasting. That is to say, if iron is immersed or "pickled" in a concentrated solution of bichromate acid and is then washed and wiped, it is rendered passive, so that it resists electro-chemical attack, whether this take the form of rust formation or the well-known plating out of copper which takes place if the chromated specimen is immersed in a dilute solution of copper sulphate. In short, the action which goes on when iron rusts is in every respect analagous to that which takes place when iron is immersed in a solution of a copper salt. In the later case, copper ions carrying positive electrostatic charges are present, iron passes into solution and assumes the electro-static charge, while copper plates out and becomes visible. When iron rusts, iron passes into solution while hydrogen "plates out." Once in solution the oxygen of the air oxidizes the iron to the insoluble form of the red hydroxide known as rust. This electrolytic action can be shown taking place by the use of a special polar indicator which has been called "ferroxyl." It follows from this that anything that will inhibit electrolytic action will act the part of a rust preventative.

To what extent the various salts of chromic acid will come into use for the treatment of boiler-feed waters and for "pickling" structural material will depend upon experiments car-

ried out on a large scale. Dr. Cushman himself is emphatic in pointing out the necessity for care and conservativeness in approaching the practical application of these purely scientific investigations. One of the modern problems in boiler practice is the rapid corrosion of boiler tubes used in connection with turbine engines. The copper which is dissolved by the action of the steam jets impinging on the bronze blades of the turbine rapidly corrodes the iron in the boilers by the electrolytic action just described. Since it has been found that the presence of bichromates in feed water will prevent this action, it seems as though the solution of this important problem has been discovered. The engineering world will eagerly await the detailed publication of Dr. Cushman's researches, as well as the results of the practical tests which are sure to follow.—"The Engineering Record."

GLAND TROUBLES WITH A STEAM TURBINE.

By Edward Russell.

The old type of steam-sealed gland is used in the majority of turbines, running or under construction, in England, whereas in America the water-seal type is being adopted in all cases. The only complaint I have heard against the latter gland came from Kimberley, South Africa, where the water is so bad that it used to fill the water space in the gland with solid deposit, but this has been since overcome by using condensed water.

The water gland appears to be by far the more economical. For instance, with a machine running on a traction load, the exhaust from the relay valve cannot be used, for the simple reason that this varies exactly with the load, giving the largest exhaust when the load is the lightest. This would be a very serious drawback in a traction station, because at the heaviest load the glands would receive very little steam, with resultant loss of two or three inches of vacuum. This is overcome in traction work by sealing the glands with live steam, which is very wasteful practice on account of the relay

exhaust being entirely lost. In a large turbine this exhaust will amount to a number of pounds in an hour. Of course, in throttle governing, such as used by Willans & Robinson, this is eliminated.

In lighting stations where the load is practically steady the relay exhaust works very well, but even here it has to be assisted with a little live steam. Vacuum has been seen to vary between three and four inches with steam from the relay valve alone; then when the load has practically dropped to zero, the steam has blown out from each gland in sufficient quantity to fill the engine room.

Then, too, I have noticed another thing. The old-style ring packings answer splendidly for machines running non-condensing, but are useless for running condensing, as the packing immediately burns up, causing no end of trouble to get the grooves clean again.

A large company in the north of England, which manufactures turbines, tried a water gland of the paddle-wheel type. It practically consisted of two grooved collars shrunk on to the shaft with a paddle wheel between them. This idea was not given a fair chance, however, the result being that the whole thing was put in the lathe, the wheel turned off to a level with the two collars, then it was grooved and turned into a steam-packed gland. When these glands were put into service the fun began. The first one went on the machine that changed from non-condensing to condensing. The temperature being kept up by the sealing steam, the collars did not follow the contraction of the spindle; consequently, they shifted on the shaft and started to rub. This added more heat, which helped to increase the expansion, resulting in showers of brass from the strips which were calked into the gland, at the same time setting up heavy vibration.

Another one went in on starting up. The machine had been well warmed up and the glands sealed to pull a vacuum, with the result that when the machine had been running a few minutes both glands fired. Anyone who has not had the misfortune to witness a gland properly fired has no idea what

a brilliant display it makes. One thing is certain: If there is any loose part in the machine it will soon make itself evident.

These cases simply show one of the drawbacks to steam-sealed glands, especially when using live steam with a gland simply shrunk on.—"Power."

MODERN APPLICATIONS OF SUPERHEATING TO MARINE STEAM BOILERS.

Paper read by ARTHUR SPYER, M. Inst. C. E., at the Engineering Conference of the Institution of Civil Engineers, June 21, 1907.

In dealing with this subject in a Conference note, it is only possible to make brief allusion to some of the interesting facts at disposal.

Information on the behavior of superheated steam as a gas is still deficient, and some recent experiments carried out by Messrs. Knoblauch and Jacob, at Munich, show that the specific heat at any given pressure is considerably higher than hitherto assumed; that it diminishes as the amount of superheat increases up to a certain point, and then rises again; and these results will influence theoretical calculations made in connection with the subject. [Similar experiments are being carried out at the National Physical Laboratory, and they confirm the German results.]

Superheaters which have been applied for marine work may be divided into two classes:

1. Those fitted in the uptake, and absorbing the heat from the gases of combustion after leaving the boiler.

2. Those placed in the boiler in such a position as to absorb heat from the gases during their passage through the boilers.

As to the first type, the largest applications for marine purposes in this country have been made by Messrs. Thomas Wilson, Sons & Co., who in 1900 fitted the *Claro* with superheaters. These superheaters consist of a series of multiple

U-tubes, with collectors at the inlet and the outlet. The result obtained in the *Claro* was so satisfactory that the system has been largely extended in the Wilson fleet, and they have at the present moment twelve vessels fitted or being fitted with superheaters.

Owing to the low gas temperature available a large superheater surface has to be provided, ranging from 40 per cent. to 50 per cent. of the boiler-heating surface, in order to give a superheat of 90 degrees to 140 degrees.

As regards results, the *Aleppo*, which has been fitted with superheaters, and runs with an average of about 150 degrees of superheat, has been carefully compared as regards coal consumption on her voyages with two sister ships, identical in other respects, running on similar service and at the same speed, but not fitted with superheaters; and it has been found that the coal consumption of the *Aleppo* is 12½ per cent. less than the sister ships over several years of service.

In the case of the *Martello*, which is fitted with Babcock & Wilcox boilers, the result has been somewhat obscured by the fact that when superheaters were fitted new cylinders were also fitted, and the cylinder proportions were considerably modified from those originally obtaining. The saving was remarkable: the vessel now uses only three of her four boilers, she runs at the same speed on the same service as she did previously, and the coal consumption has been reduced from 44 tons to 31 tons per day. The amount of superheat used in the *Martello* is 90 degrees, the surface of the three boilers is 8,055 square feet, and the superheater surface 2,000 square feet.

One of the most important questions is the condition and material of the working surfaces of the cylinders and slide valves. On this point Messrs. Wilsons' experience is of great value, as it is found that there is no trouble whatever. A very small amount of lubrication is applied internally and on the piston rods. The oil used has the specially high flash point of 700 degrees, does not form an emulsion in the feed water, and is easily removable by an ordinary mechanical filter.

As regards material of the rubbing surfaces, cast iron is invariably used; piston valves are fitted in the high and intermediate cylinders and flat valves in the low-pressure cylinders, and it is found that the wear is in many instances not so much as in ordinary cases. Moreover, there is much less trouble with the piston rods and glands than where saturated steam is used, and there is less leakage from pipe joints, etc.

As regards durability, a tube was cut from the *Claro* after five years' service, and no wear could be detected in it; and recently I had a tube cut out from a superheater which has been at work in a boiler on shore for eleven years, and failed to detect any wear in it. This tube is on the table.

The Central Marine Engine Works, of Hartlepool, have also installed several superheaters of type No. 1, one case being that of the *Inchmona*, in connection with quadruple-expansion five-crank engines, and a boiler pressure of 225 pounds per square inch.

The superheaters are fitted directly in the uptakes, below the air-heating tubes, and consist of a series of spiral coils, connected at top and bottom by cast-iron headers. The results obtained in the *Inchmona* were so encouraging that the system was repeated in the *Inchdune*, *Inchmarlo* and *Nasovia*; and it is stated that the *Inchdune* and *Inchmarlo* have been able to run whole voyages on an average consumption of 1 pound of coal per indicated horsepower hour.

A vessel called the *Teutonic* has also been fitted with superheaters, the superheater in this case consisting of a series of multiple U-tubes.

The economy directly attributable to superheating, from the data in the possession of the Central Marine Engine Company, works out to about 8 per cent.; but it should be noted that the total degree of superheat imparted to the steam is only about 70 degrees. In the experience of the Central Marine Engine Company, no ill effects have been noticed in the working of the engines due to the use of superheated steam.

As regards the superheaters of the second type—namely,

those placed in an intermediate position in the boiler—the only one which has been largely used is the Babcock-Wilcox. There are isolated cases of other systems, such as the Pielock superheater, which has been adopted in an installation of ordinary cylindrical marine boilers in France, in a vessel called the *Larence*. In this system the superheater is arranged in the water space of the boiler, and consists of a steam and watertight casing enclosing the heating tubes through a portion of their length ; the fire gases flow through the tubes of the boiler, thereby superheating the steam which passes round the portion of the tubes inside the casing.

The Babcock-Wilcox superheater is applied in conjunction with the Babcock-Wilcox marine boiler, in which the gases are passed three times through the boiler-heating surface by means of suitable baffling, and the superheater is placed between the first and second passes. By this means a gas temperature is obtained sufficient to impart a good degree of superheat with a moderate surface in the superheater.

The first tests with this superheater were made in America, on a vessel called the *J. G. Wallace*, fitted with boilers of 5,800 square feet and superheaters of 830 square feet of surface. The superheaters were placed in the first pass, but arranged so that by a simple alteration of baffling they might be cut out of the circuit altogether; thus a comparison could be made using the same boilers in the same ship, and without superheaters. The result of these trials is given below:

	Saturated steam.	Superheated steam.
Indicated horsepower	1,530	1,559
Coal per indicated horsepower, pounds	1.76	1.51
Average steam pressure, pounds per square inch	238.7	237.6
Main feed temperature, degrees Fahrenheit	192.7	195
Revolutions per minute	77	79

From this it appears that the increased coal consumption due to using saturated steam was 16½ per cent.

A curious thing occurred in these two trials. It was found that when using saturated steam the power was developed with a somewhat earlier cut-off. The reason of this is not

very clear, because the same volume of superheated steam should contain sufficient heat to develop the same power, although, of course, the weight would be very considerably less.

Following on the *J. G. Wallace*, the American Navy made an experiment by fitting superheaters to four out of the eight boilers of the *Indiana*. The results were not published, but that it was favorable is certain from the fact that since then superheaters have been ordered for four other American warships.

There are now under construction two sister ships of the *J. G. Wallace*, fitted with superheaters, and also a vessel named the *Creole*. This latter, when complete, will be of special interest, as the vessel is fitted with Curtis turbines, working at 250 pounds pressure. The principal particulars of this installation are as follow:

Indicated horsepower	7,500
Heating surface in boilers, square feet	28,150
Heating surface in superheaters, square feet	4,350
Grate area, square feet	783
Number of boilers	10

Recently the Admiralty decided to fit superheaters to six of the boilers in H. M. S. *Britannia*. A boiler fitted with superheater was erected in the test house at Messrs. Babcock & Wilcox's works at Renfrew, and four trials were made at rates of coal consumption varying from 14 pounds to 30 pounds per square foot of grate per hour; and a noticeable point in the trials was the constancy of the superheat, which varied only from 75 degrees to 91 degrees. This is of importance, as for navy purposes the rates of working vary considerably; and it is therefore necessary that the superheaters should be so designed that the superheat obtainable should be as nearly as possible constant.

A series of tests were carried out by the Admiralty on the official sea trials of the vessel. The first two trials were of special interest, as in one case six boilers were used without superheaters, and in the other case six boilers with superheaters. The results obtained are as follow:

	Six saturated steam boilers.	Six superheated steam boilers.
Indicated horsepower	3,521	3,410
Coal per indicated horsepower, pounds	2.07	1.77
Water per indicated horsepower:		
Main engines, pounds	16.92	13.63
Auxiliary engines, pounds	4.1	4.56
Total, engines, pounds	21.02	18.19
Steam pressure in boilers, pounds, per square inch	196.3	200.0
Superheat at boilers, degrees	...	92½
Superheat at engines, degrees	...	83
Feed temperature, degrees	72.0	65.6
Water per square foot of heating surface actual, pounds	5.5	5.17
Coal per square foot of grate, pounds	17.6	14.5
Temperature in uptakes, degrees	398	347

In reference to these trials a curious feature appeared. Apparently the water consumption in the auxiliary engines was increased from 4.1 pounds per indicated horsepower using saturated steam, to 4.56 pounds per indicated horsepower using superheated steam. The reason for this is not clear, as the auxiliary engines, which are most of them of the simple expansion type, are just those which should show a considerable gain in economy when using superheated steam.

A trial was also made at 7,000 horsepower with the same two groups of boilers at work, one set on the starboard engines, the other on the port engines. The engine water consumptions were 13.26 pounds per indicated horsepower with superheated steam, against 15.01 pounds with saturated steam, each engine developing 3,500 horsepower. The uptake temperature recorded was 346 degrees for the superheater boiler and 370 degrees for the saturated-steam boilers.

On the subsequent trials of the *Britannia*, at 13,000 horsepower and at her full power of the 18,000 horsepower, the boilers with superheaters showed a lower uptake temperature than those without superheaters; at 13,000 horsepower the difference was 29 degrees, and at 18,000 horsepower it was 7 degrees.

Experience, therefore, would appear to show that with reciprocating engines, when properly adapted for the use of

superheated steam, a high degree of economy is obtainable, and that no detrimental results arise from its use. There is apparently no reason, seeing that turbines can be constructed to use superheated steam satisfactorily for land purposes, why they should not also be constructed to use superheated steam for marine purposes; and developments in that direction would appear to be the next step, as already indicated in the American vessel *Creole*, with Curtis turbines.

There are, I believe, some other owners who have used superheaters for marine work, but there is, unfortunately, not any information available as to the experience obtained with them; and I can only hope that this brief indication of some of the results actually obtained with superheated steam for marine purposes may be of interest.

A NEW TORSION METER.

By B. Hopkinson and L. G. P. Thring.

With the increasing use of turbines for marine propulsion the problem of directly measuring the power delivered to a shaft by the engine has become of pressing importance. Several instruments for this purpose have been designed and have proved more or less satisfactory, the best known being that of Messrs. Denny & Johnson, which is now considerably used. All these instruments measure the twist in a length of the shaft, whence, knowing the rigidity of the shaft and the speed of revolution, the horsepower transmitted can be at once deduced. In Messrs. Denny & Johnson's torsion meter, and in the others with which the authors are acquainted, it is necessary, in order to secure reasonably accurate results, that a considerable length of shafting be taken. In Messrs. Denny & Johnson's apparatus two wheels are clamped to the shaft as far apart as is conveniently possible, and the relative displacement of these two wheels produced by twisting of the shaft is measured by an electrical device. According to figures given by Mr. Archibald Denny at the meeting of the

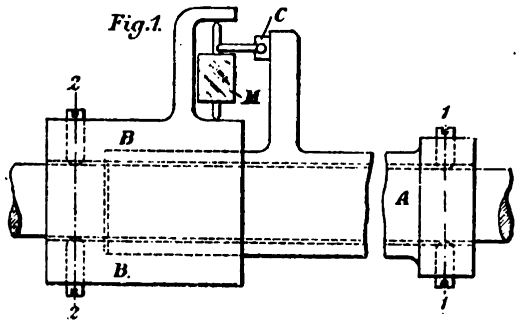
Institution of Naval Architects in March, 1907, the displacement at the periphery of the wheels can be determined correct to $\frac{1}{100}$ inch, a remarkably good result, considering the nature of the means employed. If we assume that the wheels are three times the diameter of the shaft, and that the shear in the latter at its periphery, when running fully loaded, is $\frac{1}{2000}$, the corresponding displacement at the surface of the wheels, if placed 50 feet apart, will be $3 \times \frac{50}{2000} = .075$ foot, or $\frac{9}{10}$ inch. Thus, under these circumstances, an error of $\frac{1}{100}$ inch in measuring the relative displacement of the two wheels amounts to about 1 per cent. on the full-load twist of the shaft. If a length of only 10 feet were available, the corresponding possible error would be 5 per cent. It may be assumed, therefore, that a length of the order of 10 feet is the minimum which will give good results with this form of apparatus; and Mr. Denny, at the meeting referred to, stated that they rarely made use of less than 15 feet or 20 feet of shafting. This requirement, while not militating greatly against the use of the apparatus in big ships where there is plenty of room round the shafting, and considerable lengths are easily accessible, seriously restricts its application in small vessels.

It should be observed that with a given ratio of wheel diameter to shaft diameter, the length of shafting required to secure a given percentage of accuracy is independent of the diameter of the shaft; for all shafts when fully loaded are sheared by approximately the same amount at the periphery. Thus two points a given distance apart on the surface of any shaft will be relatively displaced by a constant amount, independent of the diameter of the shaft, which may be taken as about one two-thousandth part of their distance apart. It is this relative displacement, magnified in the ratio of the wheel diameter to the shaft diameter, which is really measured by these torsion meters.

In the authors' apparatus, described in this paper, the length of shaft in which the twist is measured may be reduced to about 12 inches, the twist being measured correct to 1 per

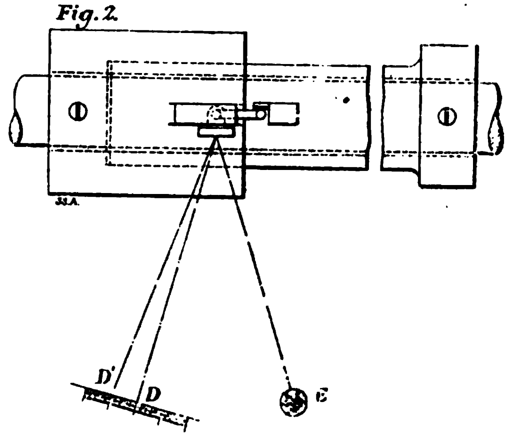
cent. of the maximum which the shaft can transmit in normal working. Figs. 1 and 2 show one of the earliest forms that were tried; it was subsequently altered considerably in detail, and is given here only because it is easy to follow the principle in it. The sleeve A is clamped to the shaft by screws

Fig. 1.



in the plane 1, 1. The collar B is similarly clamped in the plane 2, 2. On the collar is mounted a mirror M which is pivoted in a frame carried on the collar, so that it can turn about an axis at right angles to that of the shaft. The axis

Fig. 2.



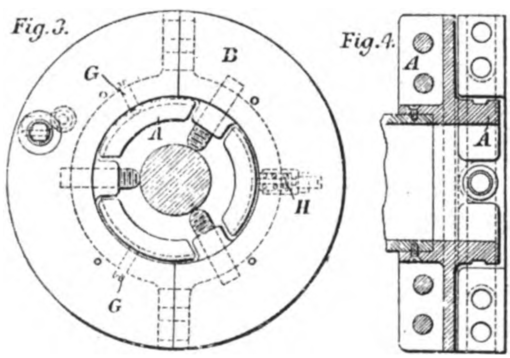
of the mirror carries a short arm which engages with a flat plate C carried on the free end of the sleeve. Twist of the shaft between the planes 1, 1 and 2, 2 causes the plate C to move relative to the supports of the mirror, thus tilting the latter about its axis. When the shaft is in the position shown,

the mirror, which is concave, forms (on a ground-glass screen) at D an image of a straight-filament lamp placed at E. When the shaft twists so that the mirror turns, this image is displaced to D′ through a distance which is proportional to the relative motion of sleeve and collar, or to the twist in the shaft between the planes 1, 1 and 2, 2. If the shaft is revolving, the image is still formed momentarily once in each revolution, and its position can easily be seen. Assuming that the distance between the planes 1, 1 and 2, 2 is 12.5 inches, and that the shear at the surface of the shaft is $\frac{1}{2000}$ (which about corresponds to normal full-load running), the relative displacement of the points of attachment of the sleeve and collar will be 0.00625 inch, and this, as already explained, will be the same whatever the diameter of the shaft. In the apparatus constructed by the authors the distance of the actuating plate C on the collar from the center of the shaft is 2.9 times the shaft radius. The movement of the plate will then be 2.9 times that of the surface of the shaft, or 1.8 hundredths of an inch. The corresponding angular displacement of the mirror will depend upon the radius of the arm; in the apparatus, as constructed, this is 0.3 inch, and the mirror is tilted through an angle of $\frac{0.018}{0.3} = 0.06$ of a radian. The angle through which the reflected beam is displaced will be double this, or 0.12 of a radian. Thus the linear displacement of the image corresponding to full-load running will be 0.12 of the distance from the screen to the mirror. Good definition can be obtained if the latter is 60 inches, giving a displacement of 7.2 inches. With an ordinary spectacle lens, silvered to form a mirror, it is quite easy to read the position of the image correct to $\frac{1}{20}$ inch, which corresponds to less than 1 per cent. of the maximum working torque for which the shaft is designed.

It is evident, therefore, that so far as magnification is concerned, the twist in a short length of shaft can be read quite sufficiently accurately by this method. It was found, however, that in the form of apparatus described above, when

used on small shafts, the bearing between sleeve and collar was apt to be strained in putting the instrument on or by bending of the shaft. Friction was thus set up, which caused error in reading, partly from the resulting deformation in the instrument, and partly because the shaft was relieved of part of the torque which was transferred to the instrument. The attempt at a complete constraint between the two members was therefore abandoned, and a partial constraint was substituted which permitted motion in certain ways, and so practically eliminated friction. At the same time the mirror and actuating plate were so disposed that relative motion of sleeve and collar other than pure twist produced no motion of the mirror. Such motions necessarily occur in ordinary working, in consequence of bending of the shaft, and, as the motion of the actuating plate is only about $\frac{1}{50}$-inch, they may cause serious errors unless eliminated. The manner in which this result has been secured will be apparent from the detailed description of the apparatus as used, which follows.

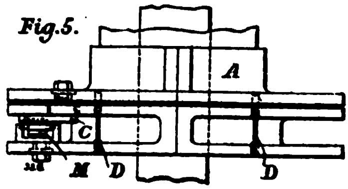
In the engravings, Figs. 3 to 5, A is the free end of the sleeve, and B the collar, each being made with deep flanges for stiffness. They are split into halves, so that the appa-



ratus can be fitted over the shaft when in place. Three screws with rounded ends are used for clamping: one set at the end of the sleeve, and another set passing through the central plane of the collar. An extension of the sleeve passes through the collar and engages with the round ends of the screws G,

G carried on the collar, being held up into engagement with them by means of a spring H. The end of the sleeve is otherwise quite free. The elasticity of the sleeve and the slight "give" in its clamping screws are sufficient to allow of its coming to a bearing on the screws G, G, without much pressure on the spring H, even when the shaft is slightly bent. The mirror is carried in a frame fixed to the flange of the

Fig. 5.



collar in such a manner that the mirror axis passes through the shaft axis. The mirror axis carries a small steel ball at its outer end, which engages with the plate C. This latter is fixed to an arm carried on the sleeve flange and passing through a hole in the collar flange, and its face is parallel to the shaft axis. The ball on the end of the mirror is approximately (in theory it should be exactly) in the plane of the screws G. With this arrangement it is obvious that any relative movement of plate and mirror, except that due to twist, will either be prevented by the bearing against the screws G, or will be innocuous, owing to its being in the plane of the plate C. Thus bending of the shaft in the plane of the mirror axis will merely cause the plate to slide over the ball at the end of the mirror arm; shortening of the shaft due to thrust has a similar effect; bending in a plane at right angles to the mirror axis is also without effect, because the ball at the end of the mirror is in the plane of the two screws G, G. When the two halves of the sleeve and collar are taken apart they are kept in their relative position by the rods D, D, which, while stiff enough for this purpose, are sufficiently elastic not to affect the reading when the apparatus is on the shaft.

The apparatus was first tested statically on a shaft of the same diameter, and probably of the same material, as the tur-

bine shaft upon which it was ultimately to be used. This shaft was $3\frac{5}{8}$ inches in diameter, and was mounted in the works of Messrs. J. I. Thornycroft & Co., Limited, with arrangements for applying a torsional load equal to about half that which it was to transmit when at work. It was found that the dynamometer reading was proportional to the load within 1 per cent. of the maximum applied throughout the whole range. The absolute twist in the shaft was also measured direct by pointers carried at the two ends. The modulus of rigidity for this shaft was thus found to be 11,800,000 pounds per square inch. There were slight signs of hysteresis, but these were not of any practical importance. The instrument was then fixed on the port wing shaft of the torpedo boat *Greenfly*, which is driven by the low-pressure turbine. The readings were taken during the Admiralty trials of the boat on March 25, 1907. On that day the boat did circling trials, and stopping and starting trials, so that the changes in power on the shaft were frequent and severe. With full steam in the turbine the shaft ran at 1,230 revolutions per minute, and the reading on the ground-glass screen, which was placed 57 inches from the mirror, was then 8.75 inches, corresponding to 1,100 horsepower. The deflection could be read easily correct to $\frac{1}{20}$ inch, or less than 1 per cent. The zero was determined by allowing the propeller to drag the turbine round with a good vacuum in the casing.

Two readings were taken, first with the boat going ahead, and, secondly, with the boat going astern, with moderate speed in each case. The mean of these readings, which differed by $\frac{4}{10}$ inch, were taken to be the zero. In the first half of the day's run the zero was found to have changed by 0.35 inch, corresponding to an error of 4 per cent. in the torque at maximum power. It is probable that this change was due to a slight shift of the scale, which was rigged up on packing cases in a very temporary fashion. Just before running home, after the trials, the zero was again carefully determined, and the boat then ran at full power for half an hour, care being taken to fix the positions of the lamp and scale. It was found

that during this run the zero had not shifted by so much as $\frac{1}{20}$ inch. Altogether the apparatus was running round for about eight hours, and was not touched in any way during that time, except once when the clamping screws were slightly tightened. It was brought down on the morning of the trial and fitted on to the shaft in about half an hour. The maximum length of shaft exposed in this boat is about 15 inches, and the clearance between the shaft and the vessel's plates is there about 8 inches; this is just sufficient to get the apparatus in. There is, of course, no independent means of checking the accuracy of the results obtained on a trial of this kind. But the facts that, when tested statically, the deflections were proportional to the twist of the shaft, and that when running the zero did not change, leave no doubt that the power readings when running must also have been correct to within 2 per cent., and probably within 1 per cent. of the maximum power transmitted.

The apparatus can be fitted on to a shaft of any size smaller than the bore, within reasonable limits; that designed for a $3\frac{1}{2}$-inch shaft works quite satisfactorily on a shaft of 2 inches diameter, and would, no doubt, work down to $1\frac{1}{2}$ inches. The over-all diameter is about $2\frac{1}{2}$ times the diameter of the largest shaft it will take. As regards length, the greater the length, of course, the greater the accuracy; but a length of 1 foot or 2 feet is ample. This length will suffice, whatever the diameter of the shaft, but the larger the shaft the easier is the accurate measurement of the power; because, while the linear displacements to be measured are the same, the disturbances due to bending and vibration become less.

The instrument, of course, measures the instantaneous value of the torque in the shaft at the moment when the reflected beam from the mirror strikes the screen. If the torque varies in the course of a revolution, as it will do with a reciprocating engine, it is necessary to measure it at several points and to take the mean. This can be done by using several mirrors placed at intervals round the apparatus, or by the use of more than one screen. In a turbine-driven shaft the torque will be

fairly uniform, though it will doubtless vary slightly with the position of the propeller blades in relation to the hull and the water surface.

Since carrying out the above-described tests we have learned that Mr. Hermann Frahm, of the firm of Messrs. Blohm & Voss, Hamburg, has been working at the same problem as that with which we have dealt, and has arrived, quite independently, though a little later in point of date, at a very similar solution. We understand that apparatus of the kind designed by Mr. Frahm has lately been tried on a German steamer, and has proved satisfactory. Mr. Frahm's apparatus is apparently the same as ours in principle, but differs from it in detail to a certain extent.

The apparatus as we have designed it can be easily applied to factory shafting, and will, we hope, prove useful for this purpose as well as in marine work, especially to those who have to put in electric motors for driving such shafting, and so require to know the exact power taken by a particular machine or set of machines.—"Engineering."

RECENT DEVELOPMENTS OF THE GASOLINE LIFEBOAT.

By George E. Walsh.

The evolution of the lifeboat has been coextensive with the growth of the life saving service of the leading nations of the world, and as no single country can claim a monopoly of patroling the coasts in the interests of humanity and property it is equally evident that the improvements in lifeboats is the result of the combined efforts of many nations instead of one or two. While the developments of new effective devices for naval fleets are held as close secrets by the different nations, and great jealousy is exhibited in protecting these improvements, the very opposite is noted in the work of increasing the efficiency of the world's life-saving service. Co-operation between the various nations is carried on extensively, and the discovery of a new plan for increasing the value of lifeboats or station equipments is quickly adopted by all.

England, America and France have led the world in the establishment of superior life-saving stations, lighthouses and other coast equipments for the protection of life and property on the high seas. Owing to their extensive and important stretches of sea coasts, it is natural that they should have taken the lead in this respect, and we look to them for the highest development of any particular feature of the work.

The lifeboat has received a greater amount of attention from these three nations than almost any other part of the service. Experiments have been conducted on a large scale by representative experts in the three countries, and while the size, shape and general construction of the boats may differ somewhat to meet local conditions, they are, in the main essentials, the same in the three countries. The need has always been of lifeboats that were not only unsinkable, but self-righting and self-bailing. A boat which could be tossed about on the waves like a cockle-shell and still preserve intact its usefulness is the only type of craft that is of any value to a life-saving station. The early type of lifeboat could often ride successfully the huge breakers on coasts, but if once capsized it was rendered useless, and likewise if filled with water by the waves it would become unmanageable, if it did not actually founder.

For some time now unsinkable lifeboats have been in use, and also boats which would automatically right themselves if capsized, and at the same time bail out the water thrown into them by the waves almost as fast as it could enter. With the lifeboat developed to such a point of efficiency there seemed but one other improvement possible. Dependent upon sail and oars for its propulsion, there were times when tide and wind would balk it from reaching a disabled ship, and repeated launchings in the surf were of no avail. If some power greater than that exerted by seamen at the oars could be installed in the lifeboats, the greatest difficulty would be overcome.

Within the last few years the problem has been solved simultaneously in England, America and France. The gasoline-

driven lifeboat is no longer an experiment, but a demonstrated actuality. Improvements in the power and range of effectiveness of the motors are still possible, but the practical use of the boats in the life saving service of the three countries indicates the future of this new type of craft.

It was difficult to install a lifeboat with a gasoline motor of the right size and capacity, and the problem involved a number of important considerations that grew more perplexing as the work proceeded. To attain anything like an ideal type it was necessary to retain present qualifications of the lifeboat, such as its ability to right itself after capsizing, bail out the water thrown into it, and be absolutely unsinkable, but at the same time possess other more intricate merits. The gasoline motor had to be placed in absolutely watertight compartments, and prove automatic in control and operation. In the event of being capsized the motor should automatically stop, and when righted again be ready to work as before.

One of the early successful adaptations of the gasoline motor to lifeboats in England was made at the Folkstone station in 1904. The experiments were tried that year with an old lifeboat, 38 feet in length and 8 feet beam. This boat was of the self-righting type, rigged with jib, fore-lug and mizzen sails, and double banks of twelve oars. She was equipped with air compartments to keep her from capsizing and sinking, but in placing a ten-horsepower motor in her some of the air cases under the deck amidships were removed. In place of the air cases a strong watertight compartment was built for the motor, pumps, vaporizers and electrical apparatus. The gasoline was carried in a metal tank forward and delivered by gravity to the engine.

This experimental craft proved so successful, especially in working to the windward of disabled ships and approaching them in a heavy sea, that a series of tests were carried on under the direction and supervision of the Royal National Lifeboat Institution. Improvements were finally made which developed a type of gasoline lifeboat which could make a speed of six knots an hour with a full crew and stores, and

when capsized by a crane the motor would automatically stop running and when released the boat would immediately right itself. When righted by its own bouyancy the motor started again by turning the starting handle. The improved life-boat was placed in regular service at New Haven, and careful study of its actions in heavy seas has since proved its value.

Another type of lifeboat driven by a gasoline motor was exhibited at the Life-Saving and Hygiene Exposition at Paris in 1904 by one who was connected with the Government Arsenal at Rochefort. This craft represented a new feature by virtue of its peculiar movable keel. In order to secure great stability in any kind of seas the lifeboats of modern times have been equipped with heavy keels fixed to the bottom. In most of the English lifeboats an iron keel weighing upward of 500 to 650 pounds was usually attached to the bottom about twenty inches below the water line. Years of practical experience had demonstrated that the weight of this keel could not be increased much more without making the craft difficult of manipulation, and it was equally impossible to extend it downward much further without fatally handicapping the boat's operation on shallow coasts.

In the Henry type of lifeboat, with its movable keel, the attempt was made to increase the boat's stability without decreasing its efficiency in either of the above particulars. The movable keel consisted of a torpedo-shaped cast-iron weight of 650 pounds. This keel was movable so that when the boat entered deep water it could be extended downward, and when approaching land the keel could be raised, or if it struck an object it would ascend of its own accord. The movable keel enters a chamber in the bottom of the boat, and when in this position the draught of the boat is only two feet, but when fully lowered the draught is increased to five feet. The result of this device is that the boat has remarkable stability in a rough sea and is most difficult to capsize. In practical tests the lifeboat, with the keel lowered, safely breasted waves which would swamp and capsize almost any other type of craft.

This same boat was equipped with sails, oars and a small motor. While the motor was primarily intended as an auxiliary power it has proved its efficiency in storms and rough weather, especially when approaching ships needing assistance. The danger of running up alongside of a ship in a heavy sea with a lifeboat propelled by oars and sail is easily imagined. With a small motor the management is made almost as simple in rough weather as in a quiet sea. The boat carrying 40 or 50 persons can make five or six miles an hour, and with only her crew and stores aboard she has developed eight miles by using the sail along with the motor.

In France the gasoline-driven lifeboat has reached a degree of efficiency that makes it of special value, and it has become an important factor in the life-saving service of that country. The first Henry lifeboat is now stationed at Brest, and others have been constructed on similar plans both in France and England. At the port of La Rochelle the experimental operations of this type of lifeboat were made a short time ago before delegates of different life-saving societies of the world.

In the United States the gasoline lifeboat has received in the last two years even more practical attention than in France and England. By virtue of our extensive sea coast, and the importance of shipping throughout its whole length, the lifeboat that can be efficiently managed in any kind of weather is of prime importance. The development of the type which has been adopted by the Government has been along lines similar to those of the English and French type, but with such modifications as local condition demand. None of the salient features of the improved lifeboat operated by oars and sails have been disregarded or weakened by the adoption of gasoline motors. The power boats are of the self-righting type, self-bailing and non-sinkable.

The American type of gasoline lifeboat embodies nearly if not all the best features of similar craft used in foreign services. The motor has been installed not alone as an auxiliary power, but as the main dependence for propulsion through rough seas to reach disabled ships. Oars and sails are also

provided, and the speed of the boat can be increased thereby, but the motor is powerful enough to secure great headway without any help from the crew.

The boat selected for the first test with gasoline-motor power has a hull 34 feet in length, 8 feet beam and 3 feet draught. It is rendered unsinkable by three cross airtight bulkheads and two longitudinal ones below the water line. These bulkheads are filled with 82 copper air tanks, so that the danger of sinking from a collision is reduced to a minimum. Half of these air tanks are sufficient to make the craft unsubmergible. To prevent the boat from capsizing there are two additional air chambers, one at the bow and the other at the stern, and these two extra chambers are sufficient to support the craft without the 82 copper tanks below the water line. The combined buoyancy of these numerous air chambers is enormous, representing upward of 12 tons. In actual tests it required 44 men to force the boat over on one side so that the deck scuppers were awash.

But if water should enter the boat, either through excessive leaning or by the breaking of huge waves over it, there is no danger of swamping. The craft is provided with abundant devices for self-bailing. A series of ten six-inch copper tubes, five on either side, run along the inside of the deck. Water shipped over the rails or when the boat is on its beam ends escapes through these tubes within a few seconds, and, as they are provided with automatic valves to shut off incoming water from the other direction, the craft can bail itself out continually as fast as the waves pour into it.

The American lifeboat is not equipped with the movable keel of the Henry type, but the placing of the heaviest weight below the center of gravity accomplishes the same end. The gunmetal keel of 1,050 pounds, the metal centerboard of 750 pounds and the copper air cases of 900 pounds weight are so placed and adjusted that the total weight of 2,700 pounds acts as a great equilibrium factor and makes the boat remarkably stable. Lateral stiffness is obtained by the large movable

centerboard, the keel and the long, flat floor. Great stability in small craft is obtained in this way without a movable keel.

The gasoline motor is of 20 horsepower, of the four-cylinder, auto-marine type, and drives an 18-inch propeller at 400 revolutions per minute. The motor is placed in the after air chamber, and is protected by a watertight door in the bulkhead. It is easy of access, but completely shut off from all air and water when in operation. Sufficient air is supplied for the purpose of completing combustion of gases by an ingenious pipe which runs direct to it from the outside. There are two fuel tanks in the forward chamber. The larger, of 75 gallons capacity, is placed in the lower part of the boat, and the smaller, of 25 gallons, above it. The fuel is pumped from the lower to the higher, and then fed to the motor from this by gravity.

The automatic control of the motor is simple and effective. When capsized the motor automatically stops, and when righted again it is immediately started by turning a crank. This control device consists of two pairs of rings mounted in a vertical position. The lower halves of the rings are of metal and the upper halves of some insulating material. A metal ball rolls freely in these rings, and makes the necessary electrical connection to cause ignition of the gas. So long as the boat is in an upright position the balls maintain constant electrical connection, but the moment the boat is upset the balls roll into the upper halves of the rings, which are composed of insulating material; electrical connections are immediately stopped and the motor ceases to operate. When the craft is righted the ball falls back into the metal halves of the rings, and it requires merely a turn of the starting crank to make the motor work.

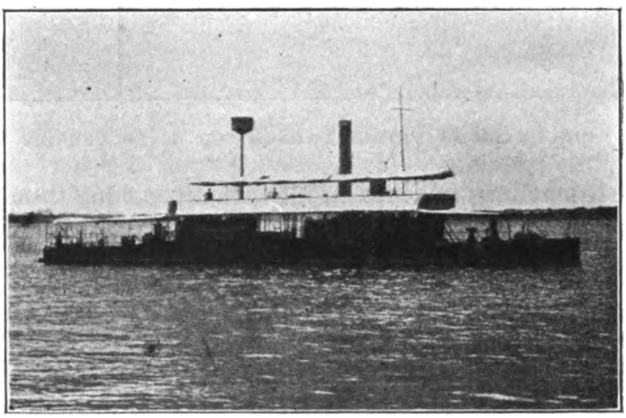
The dangers of a motor not automatically controlled when the boat is capsized in a rough sea are many. If not stopped it might get tangled in the rigging and kill some of the crew, or it would drive the boat away from the men so they could not regain it. Such control is absolutely essential to the successful operation of the modern lifeboat.

The lifeboat is steered by a rudder worked through a spur-wheel and curved rack, but it can instantly be disconnected and the rudder hoisted out of the water so that the craft can be steered by oars. Often in approaching the shore in breakers a rudder is dangerous, and steering by an oar is essential. Provision is thus made for almost any emergency.—"The American Marine Engineer."

SHIPS.

BRAZIL.

New Gunboats for the Brazilian Government.—The gunboats which we herewith illustrate are two of the seven gunboats—four of the larger and three of the smaller—recently constructed by Yarrow & Company for the Brazilian government for service on the Amazon. The dimensions of the four twin-screw shallow-draft gunboats are: Length over all, 120 feet; beam, 20 feet; depth, 4 feet 9 inches. With a load of 25 tons they draw 25 inches, and have a speed of 12½ statute miles an hour. They are fitted with Yarrow's patented hinged flap, by means of which it is claimed that one mile an hour additional speed is obtained, without additional power or additional cost.

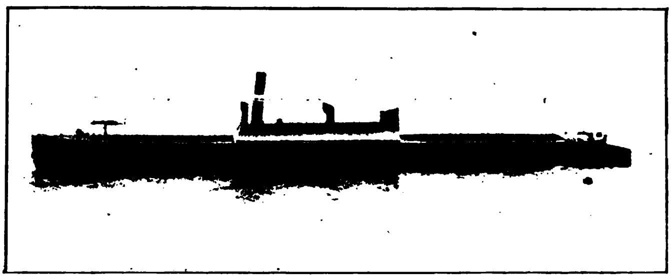


TWIN-SCREW SHALLOW-DRAUGHT BRAZILIAN GUNBOAT.

Two deck houses are provided, these being constructed of steel, one forward and one aft. The tops of these are connected by the battery deck with steel bulwarks all round,

and over this awnings extend from forward to aft, as will be seen in the engraving. The forward cabin is divided into two compartments, that forward being fitted up for the commander and the other arranged as a wardroom. The after cabin is fitted up as a petty officers' quarters, and above is provided a hollow steel mast, with crow's nest at the top. Above the forward cabin is the conning tower and the steering position, and above these again a searchlight platform.

From a structural point of view, the feature of the design is the method by which longitudinal strength is obtained for these shallow lightly-constructed vessels. It lies in the connection by the upper steel deck uniting the two deck houses, the bulwarks of this upper deck forming a girder throughout the amidship portion of the vessel.



SINGLE-SCREW PATROL LAUNCH FOR RIVER SERVICE.

The propelling engines are triple-expansion, these being supplied with steam by a Yarrow water-tube boiler. The auxiliary machinery includes steam steering engine, distilling apparatus, circulating engine, electric-light plant, fans and pumps. A steam capstan and warping gear is fitted forward, with the necessary anchor arrangements, and the usual ship's boats are provided. These vessels were shipped in pieces, and were riveted up and launched on the banks of the Amazon.

The dimensions of the three smaller single-screw gunboats are: Length, 75 feet; beam, 9 feet 3 inches; depth, 4 feet. They were also constructed on Yarrow's tunnel system, with patented adjustable flap.

As will be seen in the illustration, the machinery space is completely inclosed. The machinery and steering station are protected by chrome steel plates, proof against the Lee-Metford rifle fire point blank at 20 yards. The boats are fitted with two rudders, and can be steered either from forward or from a sheltered position aft of the machinery space. A wooden awning is provided, extending from end to end, and there is a short deck forward, on which, it will be seen, is placed a small gun.

This type of launch is now attracting considerable attention, because of its suitability for expeditionary work, its characteristic features being the extreme simplicity of the machinery, the engine being non-condensing, and the boiler of the locomotive type, which is proportioned for burning wood. The draught, with steam up, is 14 inches, and on this draught a speed of 10 statute miles an hour is easily maintained. These launches were shipped entire from London, and are considered to be just what is required to keep natives under control on some of the rivers which pass through more or less unexplored districts.

ENGLAND.

In the matter of Injuries to Ships, fate seems to be working as hard as the Government in the direction of reduction. The following is a not necessarily complete list of ships that have been ashore or badly damaged in collision during the last year or so: *Donegal*, *Good Hope*, *Dominion*, *Commonwealth*, *Montagu*, *Mars*, *Duncan*, *Albemarle*, *Prince George*.

Of these the *Good Hope* ripped up a good deal of her bottom; but as she was doing nearly 24 knots recently, her injuries must have been of a very temporary nature. The *Donegal* got off less lightly, and will probably always be something of a lame duck. The *Dominion* certainly will be, and the *Commonwealth* is likely to be non-effective for many months. The *Montagu*, of course, has gone for good. The *Mars* scraped her bottom at the Needles entrance to the Solent; no information is forthcoming as to her present con-

dition. The *Duncan* grounded at the time of the *Montagu* salving, and probably did herself no good. The *Albemarle* has been reported very seriously injured by the recent collision; but from all accounts, her name was used in error for the ship that she hit. Her real injuries are believed to be small. The *Prince George* is, or ought to be, used to accidents by now, and seems to have been unhurt by her recent hitting of a sand bank.

H. M. S. Bellerophon.—The latest first-class battleship which has been built for the British Navy, the *Bellerophon*, was launched from the Royal Naval Dockyard at Portsmouth, July 27. The *Bellerophon* is a sister ship to the *Dreadnought*, but is slightly larger. Her displacement is 18,550 tons, as compared with 17,900 of the *Dreadnought*, and she draws half a foot more water. In other respects her chief dimensions are the same as those of her sister ship, that is, her length is 490 feet, beam 82 feet and draught 27 feet. She is to be fitted with turbine engines of 23,000 horsepower, which are being built at Fairfield. It is estimated that her speed will be 21 knots. We believe that the *Bellerophon* will be similarly armed to the *Dreadnought*, with the exception that she will carry 4-inch guns to resist torpedo attack, instead of 12-pounders. The rapid progress made in the construction of the ship may be gathered from the fact that she was only laid down on December 3d, 1905, and it is hoped that she will be ready for commissioning by the end of this year.

H. M. S. Dreadnought.—The accompanying tabular statement gives the results of the last progressive speed trials of the *Dreadnought*. It was intended to make forty runs, but the new propellers were so obviously unsuitable to the conditions that the experiment was abandoned after twenty-eight runs had been completed. Using propellers on one side only, five degrees of helm kept the vessel steady on her course. It will be observed that the port engines, running alone, at 297 revolutions per minute, giving 12,500 shaft horsepower, had a shaft torque approaching that given by both sets of engines

running at 341 revolutions, giving combined 27,500 horsepower.

Both the old and new propellers are of the "cavitating" species, taking large power and giving small thrust, and the direction in which improvement may be hoped is not difficult to see. As the positions of the propellers fore and aft and athwartship cannot be altered, there only remain alterations in dimensions and forms of bosses, diameter, pitch, surface and number of blades and their thicknesses. And it is well to try propellers at deep-load draught. It is well to keep in mind that the ship was about 1,150 tons heavier than she was on her acceptance trials.

H. M. S. DREADNOUGHT.

Revs. per minute.	Speed, knots.	Torque. Inner shafts.	Torque. Wing shafts.	Shaft horsepower.	Approximate. Revs. per knot.	Approximate. Slip per cent.
			Original Trials.			
132	9.0	...	...	1,750	880	17.5
165	11.0	...	...	3,500	900	19.3
298	19.4	.6	.52	17,700	922	21.2
337	21.6	.9	.6	26,900	936	22.4
341	21.8	.98	.6	27,500	938	22.5
			Recent Trials.			
215	14.6	.45	.2	7,000	884	17.8
294	19.0	.8	.4	18,000	928	21.6
330	20.7	.93	.6	26,400	957	24.0
S 233	12.0	.66	.38	6,435	1,165	37.7
S 290	15.5	.91	.56	11,500	1,122	35.4
P 248	13.0	.68	.38	7,100	1,145	36.5
P 297	15.1	.94	.64	12,500	1,180	38.4

S = Starboard engines only.
P = Port engines only.

The Dreadnought.—*Reduced Speed at Full Draught.*—A correspondent, whose position enables him to speak with authority, writes: When the *Dreadnought* carried out her full-speed trials she did 21 knots on a draught of 26 feet. On her arrival at Villagarcia, after she had completed with stores, her draught was 30 feet. It had been calculated that the

reduction in speed on completing with stores would be a ¼ knot; instead of that the speed lost was 1½ knots. On her full draught the ship is not a 21, but a 19½-knot ship. At present the officers cannot control her at 15 knots, and, therefore, under this speed she is impossible as a station keeper.

New Type of Armored Cruiser.—It is understood that the Admiralty have decided to make provision in the next shipbuilding programme for the laying down of a new type of armored cruiser at Pembroke Dockyard during 1908–9. In size she will range between a second-class cruiser and the ships of the *Duke of Edinburgh* class. The vessel will be protected by side armor, and will mount 7.5-inch guns. She will have a speed of 23 knots per hour, and her relatively large bunker accommodation will render her, with the features above mentioned, a suitable vessel for detached police duties in remote stations.

Boadicea.—The unarmored cruiser *Boadicea*, recently laid down at Pembroke Dockyard, will have a length of 385 feet; breadth, 41 feet; load draught, 13 feet 6 inches, and displacement 3,300 tons. She will be fitted with turbine engines of 18,000 H.P. Her armament will consist of five 4-inch breech-loading guns. She will have double bottoms constructed to carry oil fuel, and a large bunker capacity.

Tartar.—The torpedo-boat destroyer *Tartar*, ordered from Messrs. John I. Thornycroft & Co., Ltd., has been successfully launched from the company's Southampton works. The dimensions of the vessel are: Length, 270 feet; beam, 26 feet; draught, 8 feet 2 inches. The contract speed is 33 knots. The vessel is to be fitted with turbine machinery of the Parsons type (built by Messrs. Thornycroft) and six Thornycroft water-tube boilers, arranged for burning oil fuel. The armament will consist of three 12-pounder quick-firing guns and two 18-inch torpedo tubes.

H. M. S. Cyclops.—This vessel has now been fitted to act as a floating dockyard. On her lowest deck is a fully-equipped foundry and forge, with cupolas, where damaged parts of machinery, can be replaced by new castings. Then there are

carpenters', blacksmiths' and armorers' shops fully equipped, fitting works, coppersmiths' and electricians' departments. On a higher deck is a boiler shop, where boiler and ship plates can be dealt with, punching and shearing machines being there just as in a land shipyard. A powerful crane travels all round the ship to lift repairs from the holds and on or off the warship that has come up for repairs. An electricity-generating station is also included in the *Cyclops*' equipment for by this power all the machines and cranes are worked. Ice-breaking plant is also carried, and, as a contrast, refrigerating machinery is also represented, whilst on another deck is a gigantic set of condensers, capable of supplying a whole fleet with fresh water. A crew of 300 men, mostly artificers, will be required on the vessel, which is believed to be the only specially-built craft of its kind in the world.

FRANCE.

French Submarines.—All the submarines actually in service in the French Navy are propelled entirely by electricity; and have but a limited radius of action. This is their principal fault. The radius of action of the earlier types only allows of a journey of from 30 to 40 miles from the electrical charging station. In addition to this the accumulators are subject to rapid deterioration; they require careful management, take up a lot of space, and are very heavy. All these added together limit the employment of this type of boat.

The speed is, as a rule, from 8 to 10 knots, and in order to remedy the slow speed and a limited radius of action, the types most recently launched or being constructed have been considerably modified. The submarine has been made more closely to approximate to the submersible, and in the near future they will form only one class. The submarines of 320 tons displacement will be provided with petrol motors of 600 H.P., capable of giving them a speed of 12 knots.

The French Submarine Gymnote recently sank as she was leaving the docks at Toulon. One is not altogether surprised at the fate which has overtaken this boat. From the very first

her behavior has been erratic, and she has met with more than one mishap previous to her last exploit. The incident is a reminder how much has still to be learnt with regard to the design and construction of submarine boats in order to insure safety and stability under service conditions. She is a single-screw steel submarine boat, designed by *Gustave Zédé*. Her length is 56 feet 5 inches; diameter, 5 feet 11 inches; displacement, 30 tons; I.H.P., 55=8 knots surface speed, 6 knots submerged speed. She carries detachable lead keel. The motive power is electricity, derived from Laurent-Cély battery. Radius, 32 miles at 8 knots, 100 miles at 4 knots. Her armament consists of two 335-mm. torpedoes, carried in drop collars, one on each beam.

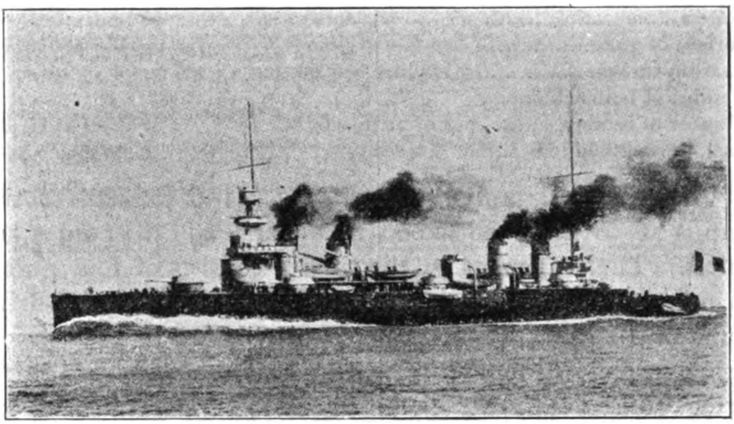
Gambetta Class.—All three cruisers of this class have now done their "reliability trials." The results are as follows: *Victor Hugo*, 96 hours—average, 19.5 knots; *Jules Ferry*, 72 hours—average, 20 knots; *Leon Gambetta*, 24 hours—average, 19 knots. At the end of twenty-four hours the *Gambetta* did a couple of hours at full speed, and made 22 knots.

The French Battleship Patrie recently burned her funnels out. According to the semi-official "Moniteur de la Flotte," Messrs. Niclausse will have to pay for this. Were a similar rule enforced in this country some boilermakers would find contracts far from profitable, as funnels burned out more or less are by no means uncommon now-a-days after hard steaming bouts.

French Cruiser Victor Hugo.—The third of the larger French armored cruisers, the *Victor Hugo*, has recently undergone her official trials. Her sister vessels are the *Leon Gambetta* and the *Jules Ferry*.

The figures which we are about to give are interesting as affording a comparison between two vessels of similar dimensions and provided with engines of the same type and calculated power, but having different types of boilers. Thus the *Leon Gambetta* had Niclausse and the *Victor Hugo* Belleville boilers. Each has the same number, namely, twenty-eight;

each has the same grate area, *i. e.*, 154 m., 1,660 feet; but whereas the *Leon Gambetta* had 55,680 square feet of heating surface, the *Victor Hugo* has 55,830 square feet. They are,



FRENCH CRUISER "VICTOR HUGO."

therefore, practically identical in every feature, though the later cruiser has a slight advantage in heating surface.

The principal particulars of these vessels are set out in the following list:

Length between perpendiculars	**148.35 m.=486 ft. 9 in.**
Breadth, extreme	**21.40 m.=70 ft 3 in.**
Displacement during trials	**12,606 tons.**
Ordinary draught	**8.10 m.=26 ft. 7 in.**
Draught, fully loaded	**8.50 m.=27 ft. 11 in.**
Total bunkers capacity	**2,100 tons.**
Contract speed for three hours	**22 knot.**
Armor:	
Fore, depth above water-line	**5.20 m.=17 ft. 1 in.**
below water-line	**1.40 m.=4 ft. 7 in.**
Midships, depth above water-line	**2.30 m.=7 ft. 7 in.**
below water line	**1.40 m.=4 ft. 7 in.**
Aft, depth above water-line	**2.30 m.=7 ft. 7 in.**
below water-line	**3 ft. 9 in.**
Thickness, fore	**90 mm.=3½ in.; 70 mm.=3 in., and 56 mm.=2 in.**
midship	**170 mm.=7 in.; and 125 mm.=5 in.**
aft	**90 mm.=3½ in.; and 70 mm.=3 in;**
Plates	**20 mm.=¾ in.**
Armor, athwartships bulkhead, fore and aft	**120 mm.=4.8 in.**

Lower armor deck, thickness, flat		40 mm.=1.8 in.
sloping		56 mm.=2.2 in.
Upper armor deck, thickness, flat		20 mm.=0.8 in.
sloping		34 mm.=1.3 in.
Engines, three in number—	*Leon Gambetta.*	*Victor Hugo.*
Boilers	28 Niclausse.	28 Belleville.
Surface of grate	154 m.e =1,660 ft.e	194 m.e =1,660 sq. ft.
Heating surface	5,173 m.e =55,680 ft.e	5,186 m.e =55,830 sq. ft.
Number of boiler rooms	8	8
Number of funnels	4 of 21 m.=69 ft.	4 of 21 m. 69 ft.
Coal, normal bunkers	1,400 tons.	1,400 tons.

With the full bunker capacity of 2,100 tons the active radius at 10 knots is 12,000 miles, and at full speed 1,500 miles. The armament in the turrets consists of four quick-firing guns of 194 mm., (7.6 inches), and twelve quick-firing guns of 164 mm., (6½ inches); in casemates, of four quick-firing guns of 164 mm., (6½ inches); besides twenty-two quick-firing guns of 47 mm., (three-pounders). There are two torpedo tubes, each of 450 mm., (18 inches). The turret armor has a thickness of 200 mm., (8 inches) for the 7.6-inch guns and 140 mm., (5½ inches), for the 6½-inch guns. The casemates have a thickness in the outside plating of 120 mm., (5 inches); in the top plate of 22 mm., (1 inch); and in floor plate of 30 mm., (1¼ inches). The fore-and-aft fire consists of two guns of 194 mm. and eight guns of 164 mm.; and the broadside fire of four guns of 194 mm. and eight of 164 mm.

The following table gives a comparison of the trials of the *Leon Gambetta* and *Victor Hugo*:

TRIALS.

Number.	Contract.	Full speed, 3 hours.		Ordinary speed, 24 hours.			Consumption trial, 6 hours.		
		Leon Gambetta.	*Victor Hugo.*	Contract.	*Leon Gambetta.*	*Victor Hugo.*	Contract.	*Leon Gambetta.*	*Victor Hugo.*
Number of boilers at work	All	28	28	All	28	28	...	4	12
Grate surface m[e]	...	154	154	...	154	154	...	2,262	12
Mean I.H.P	27,500	29,008	28,436	16,000	16,911	...	...	2,500	8,948
Consumption per IHP., kilos.	...	29,008	0,777	0,850	0,761	0,658	0,850	0,739	0,651
m² of grate, kilos	...	29,008	143	...	83	77	...	81	76
Mean speed, knots	22	23.06	22¼	...	20.37	19	...	10.30	...
Maximum of speed	...	24 02	23.10	...	...	...	...	...	...
I.H.P	...	30,500	29,048	...	...	...	...	...	...

—"The Engineer."

Launch of the French Battleship Vérité.—The Société Anonyme des Chantiers et Ateliers de la Gironde, La Bastide-Bordeaux, launched in the presence of the Minister of Marine on Tuesday, May 28th, at 6·30 A. M., from their yard, situated on the right bank of the River Garonne, the battleship *Vérité*, the principal dimensions of which are:

Length between perpendiculars 133.8 m., feet	439
Main breadth, 24.25 m., feet and inches	79–06
Depth amidships, 12.55 m., feet	41
Draught, forward, 7.98 m., feet and inches	26–02
amidships, 8.20 m., feet and inches	26–11
aft, 8.42 m., feet and inches	27–07
Area of amidship section, 179.75 sq. m., square feet	1,932
Displacement, tons	14,870

The sister ships of the *Vérité* are the *Démocratie*, in course of completion at Brest dockyard; the *Justice*, ordered from the Forges et Chantiers de la Méditerranée, La Seyne; and the *Liberté*, in course of completion with the Chantier et Ateliers de St. Nazaire. The four ships were included in the French Navy programme for 1900, and were put down in 1902, but their construction was suspended during several months, under the Pelletan Ministry, with a view to cut down expenses, and to devote all available funds and all the industrial facilities of the nation to the construction of smaller craft and submarine boats. The submarine craze is, therefore, largely responsible for the time taken in placing the four above-mentioned units on the list of ships available for active service.

The interesting feature in the operation which took place on May 28th at Bordeaux is the fact that the *Vérité* was launched 90 per cent. completed, with her engines and boilers mounted ready, her side and deck armor and her turrets fitted and in place. In one month from now she is to be inspected by the Admiralty Commission; she will then proceed to Brest under steam for her official trial runs. Her launching weight is 12,500 tons; this, we believe, is the heaviest weight

ever launched. In this the hull and armor enter for about 5,000 tons each, and the engines and boilers for about 1,000 tons. The time occupied by the vessel while gliding down the launching ways was about 50 seconds.

Owing to the local conditions of the river and to the absence of a dry dock in connection with their yard, the Chantiers de la Gironde have invariably launched their ships almost completely finished. But the *Vérité* is the largest they have built and so launched, and will be the last they will build under similar conditions. They have recently laid the keel of the *Vergniaud*, one of the six 18,350-ton turbine, 22,500 horsepower, 19-knot battleships of 1906 programme, and have in course of construction a dry dock in which to complete this ship and all others they will build in the future. The *Vérité* was built on a slipway served by two 60-ton steam travelers, 26.6 meters (87 feet 3 inches) high and 30.5 meters (100 feet) wide, which ran over the whole length.

Among their contributions to the larger fighting units of the French Navy we may mention the armored cruiser *Kléber*, 7,700 tons; the cruiser *Protet*, 4,114 tons; the torpedo-boat-carrying cruiser *Foudre*, 6,090 tons; the cruisers *Infernet*, *Chanzy*, and a large number of torpedo boats and ships for the merchant service.

The engines of the *Vérité*, built by Messrs. Schneider & Co. at their Creusot Works, are vertical, three-cylinder, compound, three in number, each driving a propeller. They are mounted in three compartments, separated from each other by longitudinal watertight bulkheads. They develop, including the accessory engines and feed pumps, a total of 18,000 horsepower at 110 revolutions. Steam distribution is by Stephenson link motion and cylindrical valves:

Diameter of high-pressure cylinder, 0.860 m., inches	$33\frac{7}{8}$
intermediate cylinder, 1.240 m., inches	49
low-pressure cylinder, 1.920 m., inches	76
Stroke, 1.150 m., inches	$45\frac{1}{4}$

The boilers are of the Belleville type, and are twenty-two in number, their principal characteristics being:

Pressure, 21 atmospheres, pounds per square inch	308
Number of stokeholds	5
Total grate area, 127 sq. m., square feet	1,365
wetted heating surface, 3,600 sq. m., square feet	38,700

The three propellers are four-bladed; the central and starboard propeller are left-handed, and the port one is right-handed. The central propeller is 4.8 meters (15 feet 9 inches) in diameter, and the two lateral ones 5 meters (16 feet 5 inches). The contract speed for the ship is 18 knots.

Her armament is to consist of four 305-millimeter (12-inch) guns, mounted in two revolving armored turrets on the center line of the ship, one forward and one aft; ten 194-millimeter (7.63-inch) guns, six of which are mounted in six armored turrets for broadside firing, and four in four armored "blockhaus;" thirteen 65-millimeter (2.56-inch) quick-firing guns distributed over the 'tween deck; and ten 47-millimeter (1.85-inch) quick-firing guns on the bridges and in the fighting tops. There are, besides, two submarine torpedo-launching tubes for 450-millimeter (17.71-inch) torpedoes. The armament will include, also, six automobile and twenty automatic mechanical torpedoes.

She is protected by belt armor formed of cemented steel plates 280 millimeters (11 inches) thick at the center, tapering down to 180 millimeters (7 inches) at both ends of the ship; also by an upper armored deck of special steel plates 54 millimeters (2⅛ inches) thick, and by a lower armored deck, the horizontal plates of which are 51 millimeters (2 inches) thick, and those on the slant 70 millimeters (2¾ inches). The armor of the rotating turrets for the 305-millimeter (12-inch) guns is 280 millimeters (11 inches) thick, formed of cemented steel plates; the 194-millimeter (7.63-inch) guns are protected by plates of the same quality, 200 millimeters (7.87 inches) in thickness; the "blockhaus" plates are 300 millimeters (11.81 inches) thick.

GERMANY.

A New Turbine-propelled Torpedo Boat known as *G 137*, built at Kiel, developed a speed of 32 knots an hour on her trial trip on July 30 in Eckernfoerde Bay.

Danzig.—This small German cruiser on her trials made 22.96 knots with 12,114 indicated horsepower. She thus fell a trifle short of her designed speed. She is the last of the seven *Bremen* class, and the only one to be under 23 knots on trial. It is interesting to note that with most of these German cruisers those built by contract beat those constructed in the Imperial Dockyards.

The trial speeds obtained by this particular *Bremen* class were as follows:

	Speed, knots.	Builder.
Bremen	23.2	Weser, Bremen.
Hamburg	23.1	Vulcan, Stettin.
Berlin	23.2	Danzig Dockyard.
Lubeck (turbines)	23.5	Vulcan, Stettin.
München	23.4	Weser, Bremen.
Leipsic	23.0	Weser, Bremen.
Danzig	22.96	Danzig Dockyard.

Ersatz Baden.—The contract for the third German *Dreadnought*, the *Ersatz Baden*, has been secured by the Krupp Germania yard at Kiel. The displacement of these ships is now stated to be 19,000 tons. The armament is sixteen 11-inch guns of 50 calibers each. Speed, about 19 knots.

Emden is to be made into a first-class torpedo station for the German Navy. It is already a secondary station, but in future it will be the chief North Sea torpedo base. Cuxhaven is being made into a Rossyth; or, rather, the Germans are doing there what the English might have done in the Humber.

The German Armored Cruiser "F" is to be built at the Weser Yard, Bremen. She is of 19,200 tons displacement; armament, twelve 11-inch; speed, 25 knots; horsepower, 50,000. Turbine machinery.

German Torpedo Boats.—A series of torpedo boats, *G 132*

to *G 137*, is now completed. These torpedo boats have a displacement of 440 tons each and steam at the rate of 27 knots per hour; one of them even exceeded 28 knots per hour on her trial trips. A new series of German torpedo boats is contemplated with a prescribed speed of 30 knots per hour.

Schlesien and Schleswig-Holstein.—With the launch of these two ships there will be twenty-four German battleships afloat, having each a displacement of from 10,000 to 13,200 tons. A new period in naval construction will now begin, in which the displacement will be increased to 18,000 tons, and the armament will be completely altered. This is expected to increase the cost of building each ship from £1,-212,500 to £1,825,000. The twenty-four battleships just mentioned constitute the total output in battleships during the present Emperor's reign. Of the number, eighteen have been built in private yards, and only six in Government yards.

THE ITALIAN NAVY.

In 1898 Messrs. C. & T. T. Pattison, of Naples, were commissioned by the Italian Government to build a number of 30-knot destroyers. The design and working drawings of the hulls and machinery were supplied by Messrs. John I. Thornycroft & Co., of Chiswick and Southampton, the boats being of a similar type to those built by them for the British and Japanese navies. The dimensions of this class are as follows: Length, 210 feet; beam, 19 feet 6 inches; draught, 7 feet 6 inches. The engines are of the four-cylinder, triple-expansion, Thornycroft, inclined, condensing type each set having four cylinders as follows: High-pressure cylinder is 22 inches in diameter, intermediate cylinder 29 inches in diameter, and two low-pressure cylinders each 30 inches diameter. The stroke is 18 inches. The contract is for 6,000 I.H.P. and the designed boiler pressure is 220 pounds.

The armament consists of one 12-pounder quick-firing gun and five 6-pounder quick-firing guns, and there are two torpedo tubes.

All the eight destroyers of this class have been launched and have fulfilled most satisfactorily the guaranteed conditions, each boat attaining a speed of at least 30 knots. It is interesting to note that they were the first destroyers to be built in Italy, and they reflect great credit on the builders, as the results obtained with them were as good as those obtained by Messrs. Thornycroft in England with boats of similar design.

Messrs. Pattison also obtained orders for four first-class torpedo boats from designs supplied by Messrs. Thornycroft, which resembled boats recently built at Chiswick for the British Government, except that twin screws were fitted instead of single screws. These boats also have gone through their trials with great success.

The following table shows the mean speeds obtained on the three-hours' official trial of six of the torpedo boats:

Pegaso, ...	25.46 knots.
Perseo, ...	25.72 knots.
Procione, ...	26.02 knots.
Pallade, ...	26.63 knots.
Cigno, ...	26.40 knots.
Cassiopea, ...	25.40 knots.

The contract speed was 25 knots.

It will be seen that this was well exceeded in every case, the *Pallade*, for example, making 26.63 knots.

Subsequently to the placing of these orders the Italian Government called for tenders from Italian builders for destroyers and torpedo boats which were to be of Thornycroft type. Messrs. Ansaldo Armstrong & Co., of Genoa, and Messrs. Odero fu Alesso, of Sestri Ponente, became licensees of Messrs. Thornycroft in Italy, and orders were placed with the former firm for four destroyers and with the latter for six torpedo boats, Messrs. Pattison also getting six additional torpedo boats.

In the design of the new destroyers Messrs. John I. Thornycroft & Co., Ltd., at the desire of the Italian Government,

arranged for complete coal protection of the engines as well as the boilers of both types of boat. In the destroyers the engines are echeloned and each engine is in a separate watertight compartment.

The leading dimensions are: Length, 211 feet 6 inches; beam, 20 feet 0 inches; draught, 7 feet 6 inches.

The engines are of the same type as those of the *Nembo* class already described, but the cylinders are 21-inch, 28½-inch and two of 30-inch diameter by 18-inch stroke. There are, as in the *Nembo* class, three Thornycroft water-tube boilers.

The armament consists of four 12-pounder quick-firing guns, and there are three torpedo tubes. The full complement of men is fifty-three in both the *Nembo* and *Bersaglieve* classes.

It may here be mentioned that Messrs. Ansaldo Armstrong & Co. are also now building six stock destroyers identical in all respects with those of the *Bersaglieve* class for the Italian Admiralty.

Twenty-four of the thirty Thornycroft water-tube boilers required for these ten destroyers have been or are being built by Messrs. Thornycroft at Chiswick, the remaining six being built, under license, by Messrs. Pattison at Naples. The leading dimensions are as follows: Length, 164 feet 0 inches; beam, 17 feet 4 inches; draught, 7 feet 0 inches.

The engines are of Thornycroft type and consist of two sets of vertical, triple-expansion condensing engines with three cylinders, 17 inches, 23½ inches and 36 inches in diameter, the stroke being 14 inches. They are capable of developing about 3,000 I.H.P. There are two Thornycroft water-tube boilers in separate watertight compartments, which supply steam at 200 pounds pressure. The armament consists of three 3-pounders and there are three torpedo tubes.

Three of these vessels have been tried and have exceeded their contract speed, the *Calliope* having attained a speed of 26.38 knots.

Trial Results of the *Regina Elena* in her 24-hours' trials

at four-fifths power showed 15,200 horsepower—I.H.P. 15,-473=20.33. The *Vittor Emanuele* in her preliminary runs at four-fifths for eight hours made 16,000 I.H.P. and 21 knots.

JAPAN.

The report that the Japanese battleships *Aki* and *Satsuma* have turbine machinery is incorrect. Both are fitted with reciprocating engines. The two new battleships in hand are, however, to have turbines. The report that one of these will be built in England is premature, because preparations for building both have been made in Japan. A sum of £2,500,-000 has, however, been allotted to replace ships lost in the late war, and it is not impossible that, Japanese yards being full, some of this money will find its way to other countries.

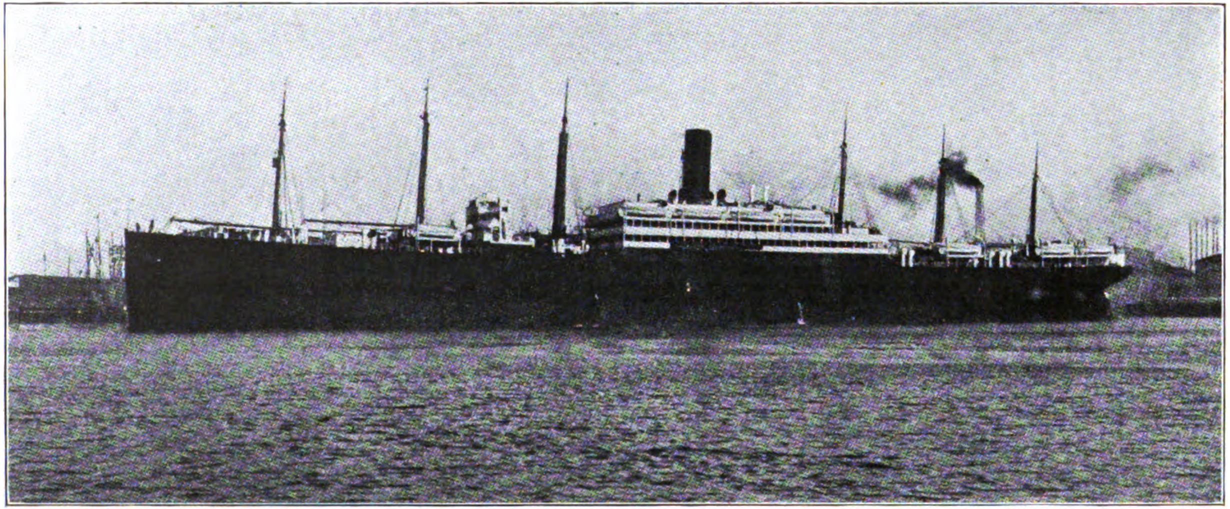
RUSSIA.

Preparations for building a *Dreadnought* have commenced in Russia at Galernii Island Dockyard. This ship will carry ten 12-inch, but be of over 22,000 tons displacement. It is estimated that she will cost well over ten million dollars.

MISCELLANEOUS.

Portugal intends adding to her fleet two destroyers, six torpedo boats and two submarines.

Two Austrian Submarines have been ordered in Germany. They will be of the same general type as "*U*," but larger, as the displacement is about 300 tons. The horsepower will be 300, and the surface speed from 12 to 13 knots.



The Hamburg-American Steamship "President Lincoln;" The First Six-Master Since the "Great Eastern."

MERCHANT SHIPS.

The New Hamburg-American Liner President Lincoln.—This new steamer, which is a sister of the *President Grant* (soon to appear), reached New York on her maiden trip, after a stormy voyage, early on the morning of June 13, and sailed for Hamburg June 22. The ship covered the distance of 3,047 nautical miles from Cuxhaven to Sandy Hook in 9 days 18 hours and 54 minutes, the average speed being 12.97 knots. The days' runs were 360 nautical miles as a maximum and 241 as a minimum, due to bad weather. The maximum represents a speed of 15 knots and the minimum of 10.04 knots.

These steamers were built by Harland & Wolff, Limited, Belfast, and were intended originally for the Leyland Line. Before completion, however, both vessels were taken over by the Hamburg-American Line. They measure 615 feet in length, with a beam of 68 feet 6 inches, a gross tonnage of 17,540 tons, and a net tonnage of 11,233 tons. The hull is constructed of steel, with a cellular double bottom extending the full length, and watertight bulkheads form a subdivision intended to provide fully for safety. The steamer presents a unique appearance, largely because of her six pole masts fitted with cargo derricks, this being the first six-masted steamer, since the famous *Great Eastern*, to enter the port of New York. The derricks are all of five tons' capacity, and six are located on each mast, with the exception of the fourth and sixth masts, counting from the bow, which have respectively four and three derricks.

On the promenade, bridge awning and upper decks are provided accommodation for 324 first-class passengers, a number of the staterooms being fitted with a single berth each. The

first-class dining saloon is situated on the upper deck amidships, and covers the full breadth of the vessel. Accommodation is provided for 228 diners. This compartment is finished in polished hardwood, with satinwood and lincrusta panels and a parquet floor. A large smoking room is placed aft on the boat deck, the paneling and framing being of carved oak, and the seats and chairs upholstered in maroon leather. The floor is covered with india rubber tiles, while a skylight with a butterfly design is placed over the center of the apartment. The social hall is located at the forward end of the boat deck, and is paneled in carved satinwood, inlaid and upholstered in moquette. An electrically-operated gymnasium and a room for electric baths and massage have been placed on the boat and bridge decks, while a small room devoted to the wireless telegraph equipment enables messages to be sent to a radius of 150 miles, and received from a somewhat greater distance, from more powerful installations.

The second-class accommodation provides for 125 passengers in two, three and four-berth rooms, located aft of the first-class quarters. The dining saloon will seat all of these passengers at once, and is furnished and decorated in white, with medallions in panels; the smoking room and social hall are also tastefully decorated. Third-class passengers are accommodated on the middle and lower decks aft, where provision is made for 1,004. The dining saloon on the upper deck aft will seat 400, and is paneled and decorated in white. A smoking room is provided for third-class passengers. Steerage accommodations are placed on the middle and lower 'tween decks, for 2,320 persons. This makes a total passenger capacity of 3,773. The crew numbers 344, making a total personnel of 4,117 when all berths are filled.

To deal expeditiously with the large cargo which this vessel will carry there are eleven cargo hatches for the seven holds, and steam winches operate in conjunction with the steel-derrick booms, before mentioned, to load and unload the cargo. On the orlop deck are large store rooms for provisions for the passengers, including refrigerating rooms for meats,

etc. The system of ventilation consists of both natural and artificial types, the latter being by means of electric fans. Electric hoists are installed to convey goods from the store rooms to the galleys and pantries, and a number of electrically-operated machines are in use in the culinary department for various purposes. There are sixteen large lifeboats, besides a provision of life rafts carried in four stacks. The ship is fitted with bilge keels, to minimize rolling.

The propelling machinery consists of two sets of quadruple-expansion engines driving twin screws, with a maximum of 8,500 horsepower and a sea speed of 14 knots. The cylinder diameters are respectively 25, 36, 52 and 75 inches, with a common stroke of 54 inches. At full power the revolutions per minute are 86, corresponding with a piston speed of 774 feet per minute. The low-pressure cylinders are fitted with D-slide valves and a balance cylinder, the other cylinders having piston valves, one for each cylinder. The arrangement of cylinders places the high-pressure forward, with the second intermediate next, then the low-pressure, and the first intermediate aft. The high-pressure and low-pressure valves are forward of their respective cylinders, the other valves being aft. These valves are operated by Stephenson double-bar links, actuated by two eccentrics each. The engines are balanced on the Yarrow-Schlick-Tweedy system.

Steam is furnished by six boilers, of which four are double-ended and two single-ended. There are three furnaces in each end of the double-ended boilers and three in each of the single-ended, making a total of thirty furnaces. The aggregate grate surface is 55 square meters (592 square feet); the heating surface is 2,100 square meters (22,600 square feet); the ratio between the two is 38.2 to 1. The boilers are all of a diameter of 14 feet 8 inches, the lengths being 19 feet 10 inches and 10 feet 9 inches respectively. Howden's forced-draft installation is fitted. The steam pressure is 215 pounds per square inch.

The condensers are located in the back columns of the low and first intermediate cylinders, that is, in the after half of

the engines. The propellers have each a diameter of 6 meters (19 feet 8 inches) and a pitch of 6.1 meters (20 feet); this makes a pitch ratio of 1.017. The shafting is carried out through spectacle frames to avoid the use of outside struts. There are eleven sections in each shaft, the two sections of crank shaft having a diameter of 15½ inches. The thrust shaft has seven thrust collars. The line shaft, in seven sections, is provided with two bearings to each section for steadying purposes.

Four sets of direct-connected generators and engines furnish electricity for both lighting and power purposes. Three of these are of 480 ampères capacity each, and the other of 200 ampères; all are operated at 100 volts, the larger types being run at 300 revolutions per minute. The three larger ones have compound engines with cylinders 10 and 17 by 10 inches, and are located just aft of the engines on the main engine-room floor. The fourth unit has a single cylinder, 9 by 8 inches, and is located in a little compartment above the water line, just aft of the engine room, in order to be available in case of emergency, should the engine room be flooded. All were furnished by W. H. Allen, Sons & Company, Bedford. The refrigerating machinery, which includes a compound steam engine operating a single compressor, was furnished by J. & E. Hall, of Dartford. The anchor winch is from John H. Wilson & Company, Limited, Sandhills, Liverpool.—"International Marine Engineering."

YACHTS.

Alexandra.—The new turbine-yacht, the *Alexandra*, built for His Majesty the King, was launched on Thursday, May 30th. This vessel was the subject of competition on the part of a number of the principal yacht-builders in the United Kingdom, who were required to send designs and models to the Admiralty under conditions which precluded any possibility of the author of the design becoming known to the adjudicators. The plans were considered by a Committee, and after long consideration the order for the new yacht was placed with Messrs. J. & A. Inglis, Limited, of Glasgow. This was regarded as well-merited recognition of the undoubted reputation of the firm for their excellence in this department of naval architecture. The result will be looked forward to with considerable interest.

The new yacht is to serve as an auxiliary, rather than as a substitute, for the *Victoria* and *Albert*, built in 1899, but there can be no doubt that, owing largely to the more convenient size, and especially the lighter draught, the new vessel will be preferred for most purposes. The *Victoria* and *Albert* has a length of 380 feet, a beam of 50 feet, and her designed draught was 18 feet, equal to a displacement of 4,700 tons. The new yacht is 275 feet long, of 40 feet beam, and 12 feet 6 inches draught, and she is expected to displace 2,050 tons. She has a bridge-deck 150 feet long, which is carried over the full width of the ship, being supported at the sides on stanchions, so that there is afforded a sheltered promenade. The pavilion covered by this deck contains the reception room, dining room and pantry, with two small boudoirs arranged to afford a clear view ahead and astern as well as on each broadside. Forward under the bridge there is the smoking room, along with the rooms of the commander of the ship, and the surgeon and other officers. On the main deck in the after part of the ship is the suite of royal rooms, with rooms for the secreta-

ries, equerries, ladies-in-waiting, and other attendants. The yacht is also most admirably arranged for the accommodation of guests, officers and crew.

A feature is the adoption of the Parsons turbines for the propulsion of the ship. The turbines are designed to develop 4,500 horsepower, to give a speed of 18¼ knots. The usual three-shaft arrangement—one now universal in the mercantile marine—is adopted, with one high-pressure and two low-pressure turbines, and reversing turbines incorporated in the exhaust casing of each of the low-pressure turbines. The turbines have been designed according to the latest practice in the light of experience obtained. Special attention has been given to accessibility and to convenience in the overhaul of the various parts with a minimum of disturbance of the remainder of the engines. All the bearings are under forced lubrication, with all the latest fittings and sight-connections for the observation of free flow of oil through the bearings. None of the bearings is water jacketed, as in earlier turbines; a special independent oil cooler is provided for cooling the oil. With a view of maintaining a high vacuum the Parsons patent vacuum augmentor arrangements have been fitted in this vessel.

The starting platform and operating valves are on a level with the turbines, so that the engineer has a full view of the whole of the engine room and auxiliaries under his charge. For starting, there is only one simple valve. The maneuvering valves—*i. e.*, one for the low-pressure ahead and the other for the astern—are combined, and operated by one handle. The starting gear is arranged so that these valves are practically horizontal.

NOTES OF THE SOCIETY.

At a special meeting of the Council held on June 25, 1907, the Secretary-Treasurer was authorized to purchase one one-thousand dollar bond of The Washington Railway and Electric Company. This bond has since been obtained.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XIX. NOVEMBER, 1907. No. 4.

ENGINE EFFICIENCY AND EFFECTIVE TURNING MOMENTS AND THEIR EXPERIMENTAL DETERMINATION.

Read before the Schiffbautechnische Gesellschaft, Berlin, by
HERM. FÖTTINGER.

Translated by ENSIGN CARL A. RICHTER, U. S. N., Member.

The exact determination of the effective performance of marine engines, as soon as it dealt with amounts of several hundred horsepower, was impossible up to a short time ago. The difficulties of a brake, called forth by high powers, the constricted spaces, the inaccessibility of the engines, of the propellers, as also of the complete lack of other accurate and also practicable methods of measurement, were the reasons why we had to be satisfied with the indicated performance of the engines.

Still more meager was our information concerning the size, efficiency and course of shaft performances requiring certain

turning moments, and up to now our ideas on these points stand in considerable contrast to the actual performance.

The interesting speed measurements of Dr. Bauer, Stettin, published three years ago, and the more recent torsion measurements of Frahm, Hamburg, and Professor Denton, Hoboken, first brought light in the dark and, unfortunately, often useless, department.

Meanwhile has also the respective theory, partly outstripping experiment, experienced an improvement which nearly all actual performances bear out. I have in mind the work of Professors Lorenz, Gumbel and Frahm.

The methods of measurement and the apparatus which are here discussed owe their origin to my resolution to broaden the practical technical side of this subject, whose imperfection was, in connection with the corresponding propeller computations, disagreeably inconvenient to me. I set before myself the problem of constructing an indicator whose effective performance permitted of determining the actual turning power from a self-produced diagram. The practically obtained solution of the problem is presented in the Torsion Indicator. It is self evident that the road to the solution led through a series of complicated measuring arrangements which in turn unite other advantages.

Before we enter upon the individual methods let us touch lightly upon the difference between the effective and the indicated engine performances or H.P. Indicated H.P. is the work (energy absorbed from the steam by the pistons), which may be easily calculated from the indicator diagram. Like a stream, it flows through the connecting rods, the crank and line shafting to the propeller, its destination. At every bearing along the way a portion of the energy stream trickles away, and of the original stream of energy, the effective I.H.P., only a part is utilized in actually turning the propeller. The numerous losses are indicated in Fig. 1, of which a description would be superfluous. I only wish to emphasize the losses produced by the propellers or the engines in the ship's vibrations, which up to this time have

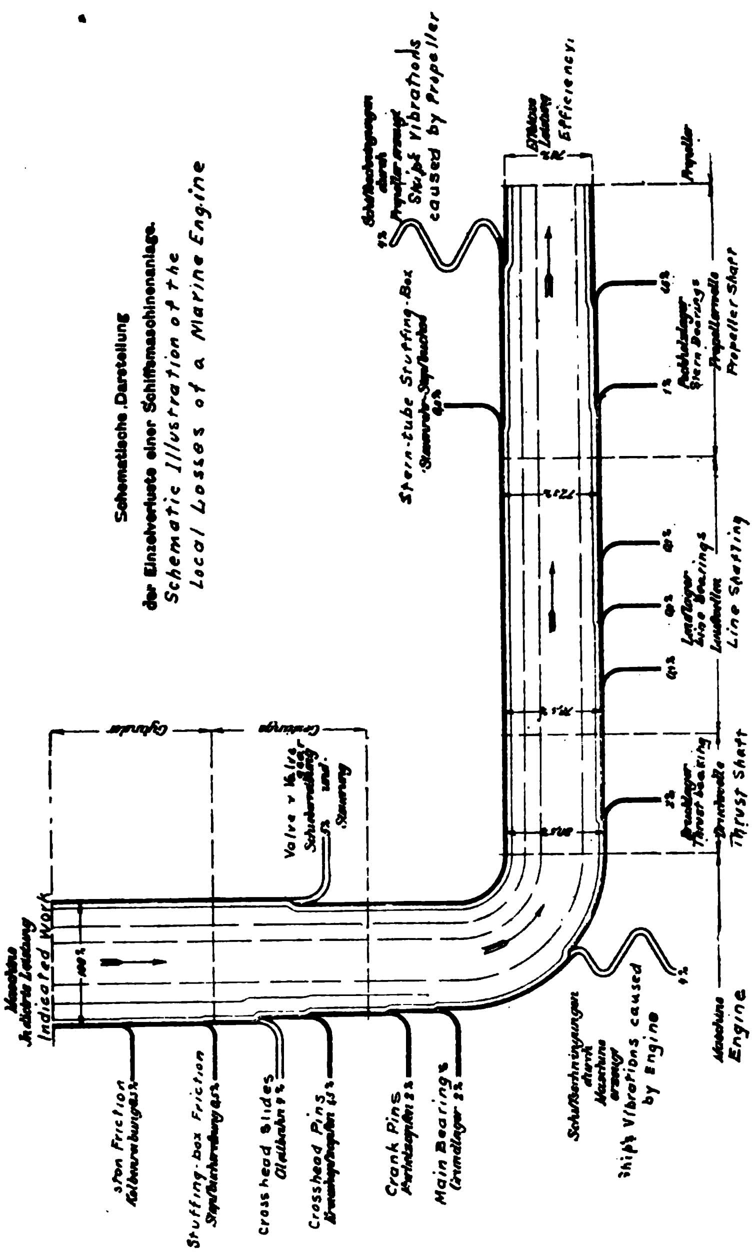

Schematische Darstellung
der Einzelverluste einer Schiffsmaschinenanlage.
Schematic Illustration of the
Local Losses of a Marine Engine
Indicated Work
Piston Friction
Stuffing-box Friction
Crosshead Slides
Crosshead Pins
Crank Pins
Main Bearing
Valve & Valve Gear
Ship's Vibrations caused by Engine
Engine
Thrust Bearing
Thrust Shaft
Line Bearings
Line Shafting
Stern-tube Stuffing-Box
Stern Bearings
Propeller Shaft
Ship's Vibrations caused by Propeller
Efficiency
Propeller

Fig. 1.

rarely been mentioned, and for whose determination I give an exact method further on.

A universally applicable definition of "efficiency" is not possible. We understand by that term the efficiency of the shaft abaft the engine, or that beyond the thrust shaft, or the stern tube, or, finally, that, up to the propeller. The same arbitrariness holds good in relation to the ratio of effective to indicated performance, which is called the "mechanical efficiency."

It would be valuable in both the scientific and the practical way if the "Schiffbautechnische Gesellschaft" (Society of Naval Architects) would follow the example of the Society of German Electricians and standardize their efficiency terms. It might be recommended that the efficiency up to the stern tube be called the efficiency, because it is the easiest to measure accurately.

Regarding the knowledge of efficiency in specifying the engine power of new ships, in calculating the best propeller sizes, and especially the shaft dimensions, there has been no lack of attempts to utilize the experimental data on hand. For this the ideal way would be the calculation of the individual or local losses, a method which is so successfully

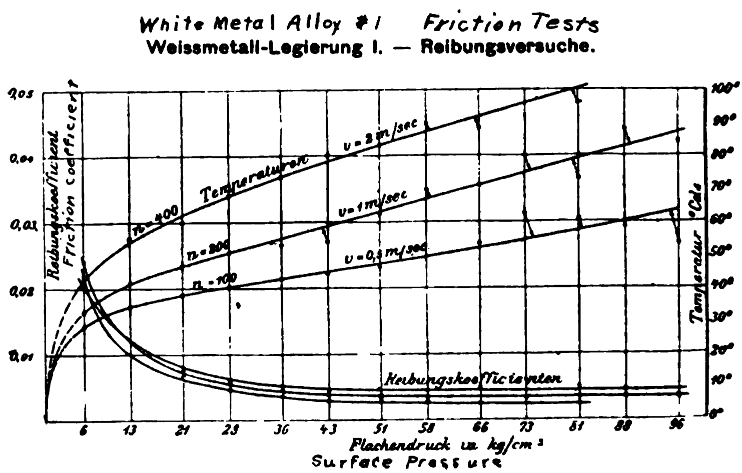


Fig. 2.

Weissmetall-Legierung II. — Reibungsversuche.

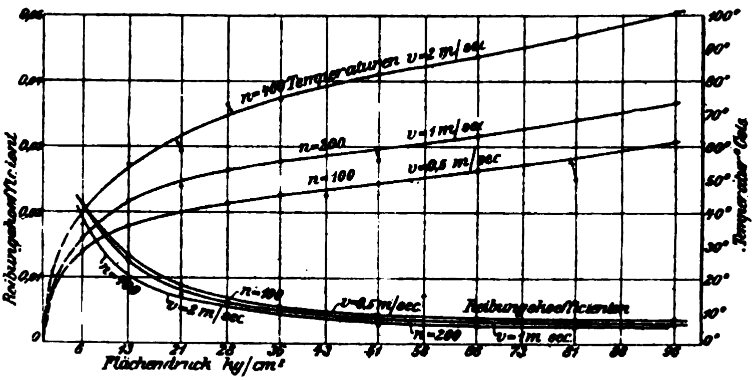


Fig. 3.

applied in the case of modern dynamos and motors that we can calculate beforehand as close as ½ per cent. the probable efficiency of larger types.

In the case of marine engines the usual way is to calculate the individual friction from the total pressure at the respective parts and from the friction coefficient. Until now the friction coefficients for bearings and the like were taken from the special experiments in which the direction and the amount of pressure were kept constant. Figs. 2 and 3 show, for example, the very great decrease in friction coefficients at high pressure with modern white-metal bearings and steel shafts.

Unfortunately, however, the journals of a marine engine work under entirely different conditions. Because of the alternating strokes the journals are lifted from the bearings twice in each revolution, and the oil is drawn in from the oil ducts, and varying conditions of oiling and friction undoubtedly enter largely into the question. The lack of knowledge of what really happens in the bearings, modern valves and stuffing boxes, and of the energy absorbed in the ship's vibrations, makes the preceding method practically worthless.

Of more importance is the calculation of the work done under no load. For stationary engines, this formula is of use:

$$N_i = N_l + \delta . N_e + N_e,$$

where

N_i = I.H.P.;
N_l = work done under no load;
N_e = effective H.P.;
δ = a constant.

For every I.H.P., N_i, there is a corresponding effective H.P., N_e, in the form

$$N_e = \frac{N_i - N_l}{1 + \delta},$$

and that also, with constant revolutions and alternating stroke.

In the case of marine engines, however, each stroke corresponds to a different revolution speed. We have at hand rough assumptions as to the relation between power and revolutions, and have attempted to apply this relation to the marine engine. The formulas obtained in this way are highly complicated and lack, at present, experimental proof in the case of modern marine engines.

An interesting way by which we may obtain an insight into the rate of work with alternating strokes has been devised by French engineers. Working on one shaft are two engines independent of each other, as is the case with some American and French warships, and the S. S. *Kaiser Wilhelm II*, and these are run so as to obtain the same speed with both or with either one. In the first case, where each engine did half the work, the I.H.P. per screw was found to be greater than in the second. In this way the above mentioned formulas were tested.

This method is generally possible. The same number of revolutions can be split in any desired way between the forward and the after engines; in fact, we can go so far as to work one engine backwards—that is, to oppose one engine to the other so that only the difference of their turning moments will be available at the propeller. In principle this experiment is possible, even to the extent of applying the full power

of the one engine against that of the other, with the propellers uncoupled. The excess of power of one engine over the other necessary to turn the shaft would give the I.H.P. necessary to overcome the combined friction of both engines at full power; that is, we would obtain directly the indicated sum total of the losses, and thereby the effective H.P.

The extraordinary danger of this last operation would forbid such a highly interesting experiment being attempted.

The oldest experimental method of obtaining the effective H.P. is the brake.

As is well known, the power developed is directly measured by a scale and the energy is transformed into heat.

Prony'scher Zaum
Prony Bridle

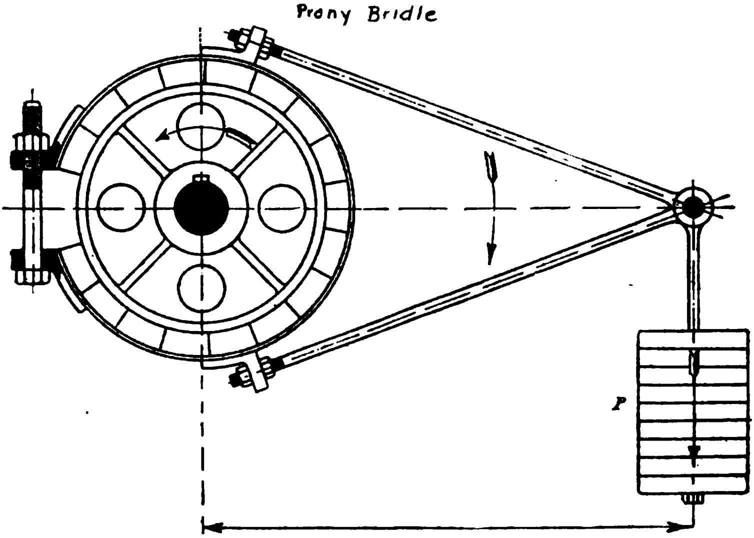


Fig. 4.

The most primitive form of brake, the Prony bridle (Fig. 4), I may assume to be well known, the principal disadvantages of which, the irregular friction and the considerable wear on the bridle, are cleverly avoided in the brake devised by Siemens and Halske (Fig. 5). At the first tendency to burn, the tension is automatically lessened by the working of an inner lever.

Adjustable Brake with Automatic Heat Preventer
Regulirbare Bandbremse mit selbstthätiger Verhütung des Festbrennens.

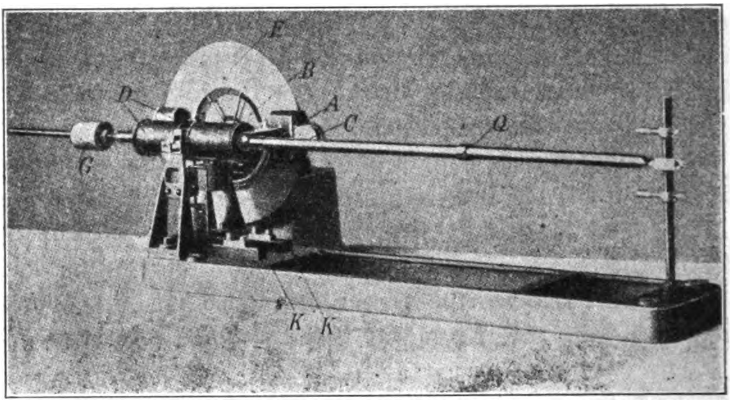


Fig. 5.

The question is prettily solved other ways by the eddy and the hydraulic brakes. The "eddy" brake substitutes for the mechanical friction the electrical action between the armature and field of a dynamo (Fig. 6). There is in a dynamo a

Schematische Darstellung der Bremswirkung einer Wechselstrommaschine.
Diagram of the Working of an Electric Brake.

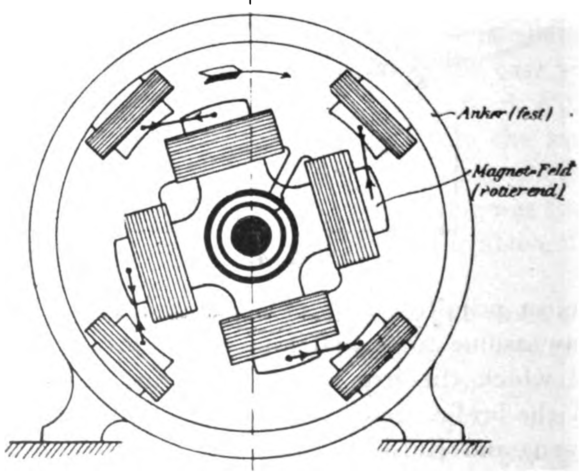


Fig. 6.

strong resistance to turning as soon as the circuit is closed. If the field were free to revolve about the armature it would soon take up the same number of revolutions as the armature; this turning of the field is prevented by a weighted lever, and the turning moment or B.H.P. is thereby measured.

This idea is worked out in the electric brakes of Siemens and Halske (Fig. 7) and of E. H. Rieter (Figs. 8–10).

Eddy Brake (Large Model)

Wirbelstrombremse (grosses Modell).

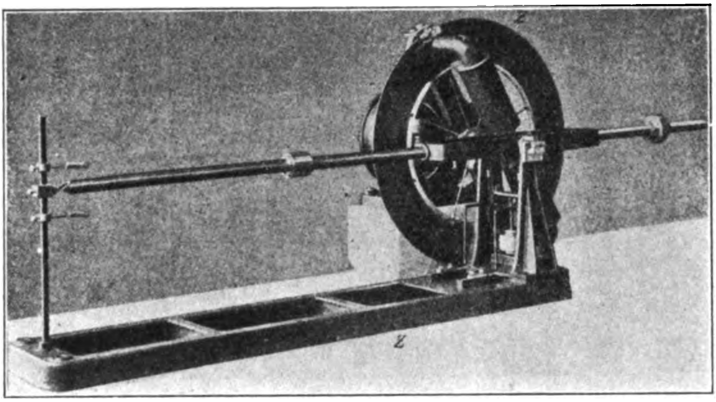


Fig. 7.

The former has a rotating copper disc, the latter an iron drum with cooling ribs, and they both have stationary magnets. A special covering or insulation for the field is not necessary, for the current generated is better dissipated in eddies on the metallic surface of the armature. Any desired number of revolutions is obtained by altering the strength of the field current. The figures show clearly the levers for measuring the torque.

Froude's hydraulic dynamometer absorbs the power by means of violent water eddies. A water wheel rotates in an enclosed water-filled casing. The wheel and the casing have corresponding annular channels of semi-elliptical cross section, the channels being divided into small cups by diagonal partitions. When the shaft rotates the water in the wheel cups

Exact Brake Dynamometer
Präcisions-Bremsdynamometer von C. H. Rieter.

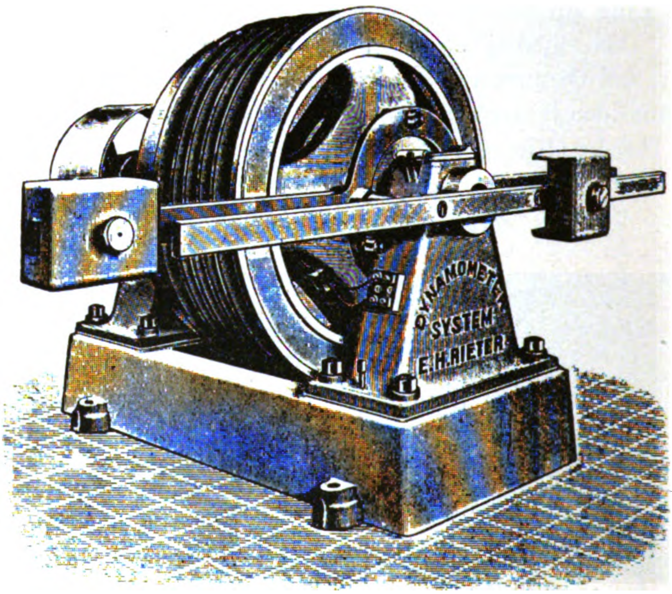


Fig. 8.

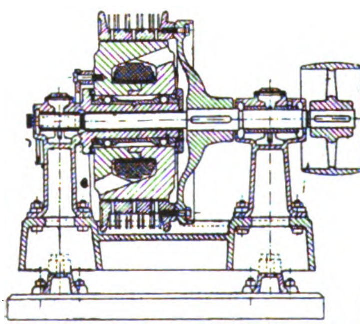


Fig. 9.

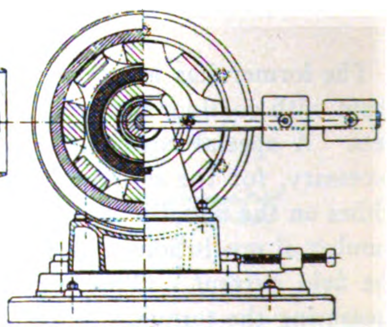


Fig. 10.

is thrown outward by centrifugal force, is sucked on near the shaft and is so forced on with increasing agitation.

The casing is movable about the shaft, and the action of the water makes it tend to rotate with the wheel. This tendency is balanced by a weighted lever, and the torque

thereby measured. The clearance spaces are adjusted by distance pieces on the horizontal screw studs. The firm of Heenan & Froude, Birmingham, advertise a water dynamometer with a 770-mm. water wheel, which will measure 1,100 B.H.P. at 350 revolutions, which is about the power of a small torpedo boat's engines.

A brake necessitates expensive and detailed preparations and measures only the average torque, while the widely vary-

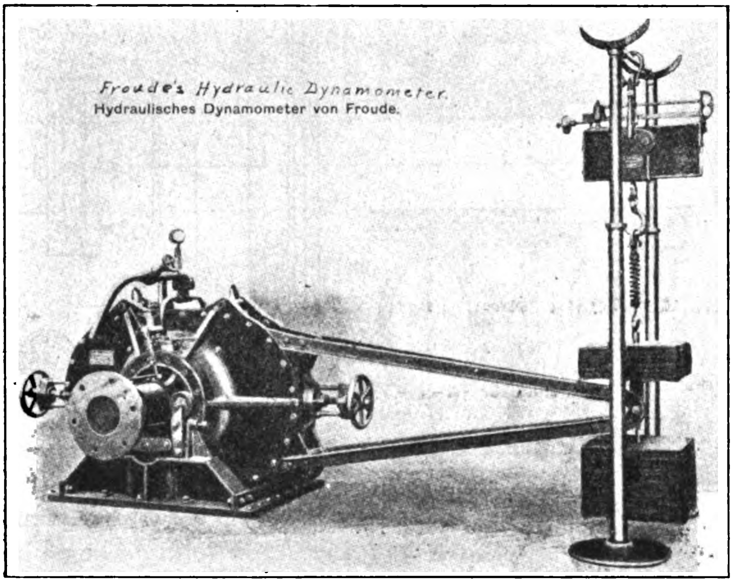
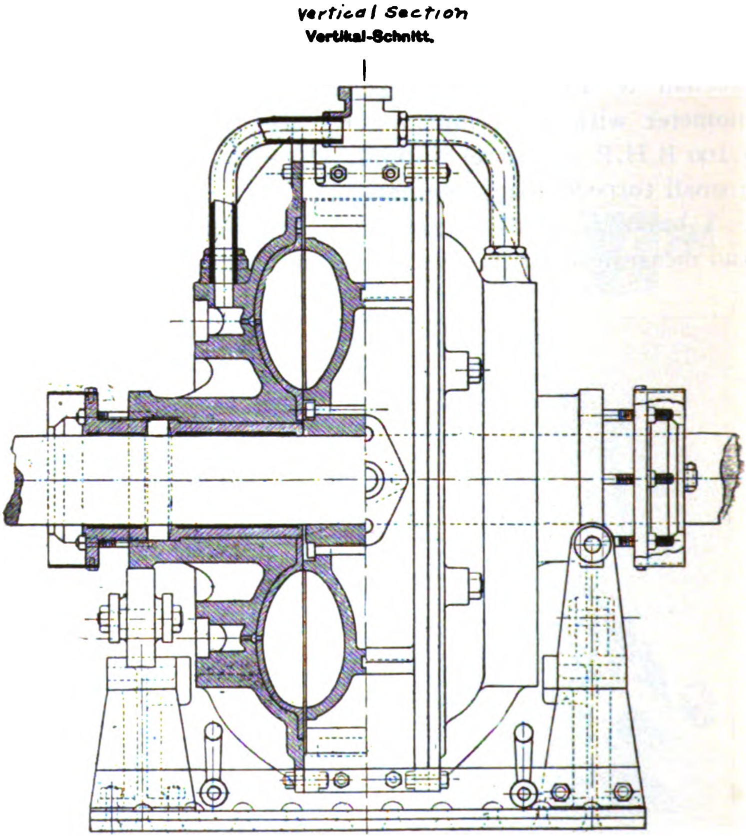


Fig. 11.

ing torques, as found in marine engines, are only qualitatively indicated by a brake in the form of oscillations.

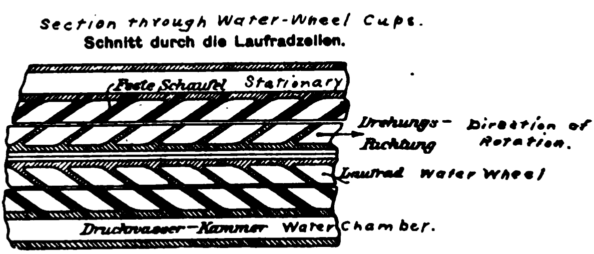
So far, in order to measure and register the variations in torque during a revolution, we had to have recourse to an inserted dynamometer. In this the shaft is broken up, and an arrangement devised which, by means of weights or springs, determines the torque.

Of the many devices of this type there are only two which are applicable to marine engines. White's cogwheel dyna-



Vertical Section
Vertikal-Schnitt.

Fig. 12.



Section through Water-Wheel Cups.
Schnitt durch die Laufradzellen.
Feste Schaufel Stationary
Drehungs-Richtung
Direction of Rotation.
Laufrad Water Wheel
Druckwasser-Kammer Water Chamber.

Fig. 13.

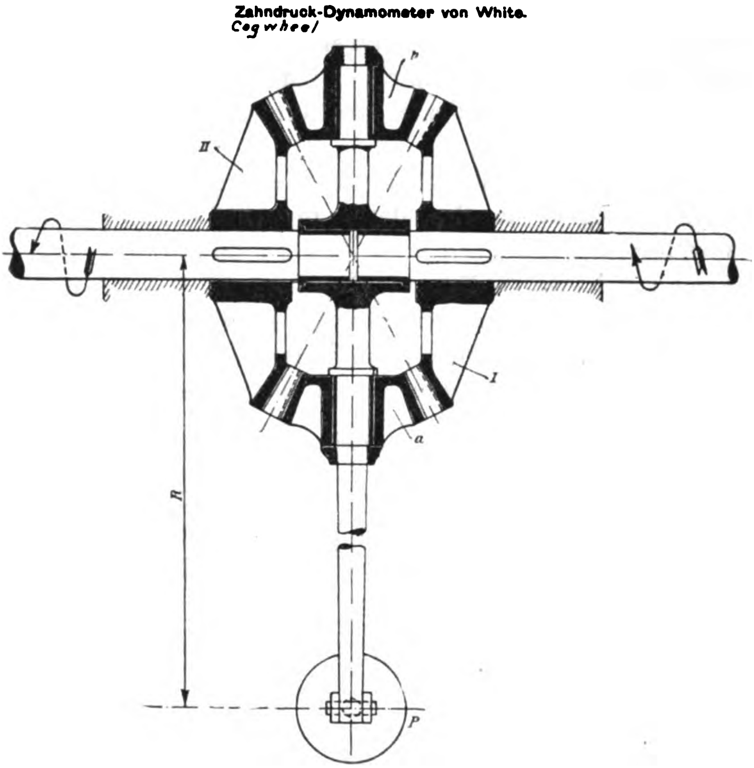


Fig. 14.

mometer is shown in Fig. 14. The entire wheels *a* and *b* are loose on an arm, which is free to revolve on the shaft and transmit the torque from *I* to *II*. It is plain that the arm must be kept from rotating, otherwise it will not transmit any power from *I* to *II*. A turning moment of $P \times E$, equal to that of the shaft, is therefore necessary to hold the arm stationary.

A weighted lever would not be applicable, on account of the noise and jolting of the spur wheels.

By replacing the weights with springs the instrument is made self-adjusting, but with this there are difficulties in dealing with more than 100 H.P., because the direction of revolution of the shaft is reversed.

The torsion dynamometer shown in Fig. 15 acts on an entirely different principle. Power is carried from one shaft to the other by means of spiral or flat springs, whose distortion at any moment indicates the torque, as is done in the case of the indicator.

The principal difficulty of this kind of apparatus lies in observing and registering the relative torque of the two sections of shafting.

Ayrton and Perry's, Fig. 15, shows a small lever K with a shining knob at the end, which turns outward as the torsion increases, and a fixed rule permits us to easily, if not very accurately, measure the diameter of the circle swept by the

Torsionsdynamometer.

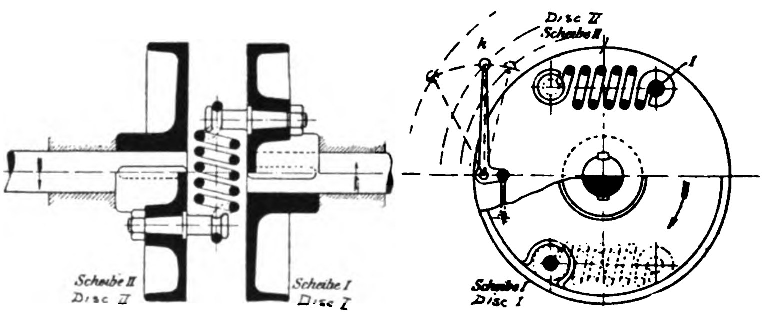


Fig. 15.

knob and thence the torque. With this apparatus the indication of the torque during one revolution is not possible. We will find the same difficulty in the case of the torsion meter, but it has there been overcome in a simple manner.

Although the inserted dynamometers give much better results than brakes, the value of the results is not at all proportionate to the expense and the danger. A broken spring means a broken shaft.

There is known only one case in marine engine practice, that of the *Turbinia*, where an inserted dynamometer is used.

In the determination of the torque during one revolution this kind of dynamometer gives false results. The experi-

mental researches of Frahm, as well as my own (Föttinger) demonstrate irrefutably, and in accordance with theory, that all the torques applied to the shaft are there affected by the torsional vibrations brought on by the crank and the propeller.

The insertion of the elastic dynamometer alters the stiffness and thereby the torsions of the shaft, so that the character of the normal impulse is entirely changed, assuming that the dynamometer is not normally inserted.

Since we are not permitted to change the nature of the shaft, there remains only one applicable method, that of obtaining the torque from the torsion of the shaft, a method that was introduced sixty years ago by the distinguished student of thermo-dynamics, Hirn, and in later years by Denton, Frahm and myself (Föttinger).

For all kinds of forged iron and steel there is a definite relation between the actual torque and the angle of torsion. The following equation holds good:

$$S = \frac{L \times R \times M}{G \times \theta} = \text{const.} \times M,$$

where

S = the measured arc of torsion in Cm,

R = radius of arc S also in Cm,

L = length in Cm of shaft over which the torsion is measured,

G = modulus of elasticity of the shaft in Kg/cm^2,

θ = polar radius of gyration of the cross section of the shaft in cm^4, and

M = actual torque in cm/Kg.

Therefore, if we measure the circumferential torsion S, we need only multiply by a constant to find the effective turning moment. We must know, of course, the modulus G, which may be accurately ascertained from experimental shafts or from the actual shaft.

Following is an example of measured torque:

With a shaft 25.2 meters long, 320 mm. diameter, transmitting 2,000 H.P. at 76 revolutions, the mean arc of torsion 315 mm. from the center of the shaft was 17.5 mm.

To obtain the maximum arc of torsion the radius R of measurement, at the length L, must be as large as possible. While it is easy to measure the torsion of a stationary laboratory shaft with aid of a magnifying glass, there are certain difficulties in the case of a rotating shaft.

PRINCIPLE OF ALL TORSION MEASUREMENTS.

The principle of all torsion measurements is shown in Fig. 16. On the two discs a and b, are scribed two radii which are parallel when the shaft is not subject to torque. When torsion exists, these radii form an angle with each other. For example, when the radius b has just reached a vertical posi-

Diagram of Torsion Measurement
Schematische Darstellung der Torsionsmessung.

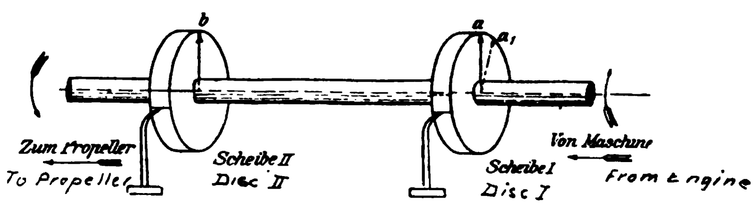


Fig. 16.

tion, the radius a has already passed beyond the vertical to the position A_1; for the rear disc is held back by the resistance of the propeller. That is, while the shaft is doing work the forward disc is always ahead of the after one by the angle of torsion.

We now assume that with a ship's shaft at rest we have scribed a zero mark on the forward and one on the after disc, and that it is possible at any instant, with the same scribing arrangement, to scribe two new marks while the shaft is working.

If the shaft were perfectly rigid and rotated without torsion

the new marks would necessarily be an equal distance from their respective zero marks.

Actually, however, the zero mark on the forward disc precedes that on the after disc by the length of torsion arc, so that the new marks will be found unequally distant from their respective zero marks by the length of torsion arc.

If we subtract, then, from each other the distances of the new marks we will have the arc of torsion at that instant. This differential principle underlies all methods of torsion measurement.

The following developments of this principle differ only in the way by which the marks are obtained.

METHOD OF SYNCHRONOUS TUNING FORKS.

This method was used by Dr. Bauer in his speed measurements. Two paper-covered drums were fixed to the shaft at a known distance apart. Two electro-magnetic tuning forks vibrated by the same current scribed wave lines on the paper. The speed of the shaft was determined from the length of the waves. By means of a simple arrangement this method may be used for torsion measurements. The two tuning forks may be made synchronous, keep step with each other, by

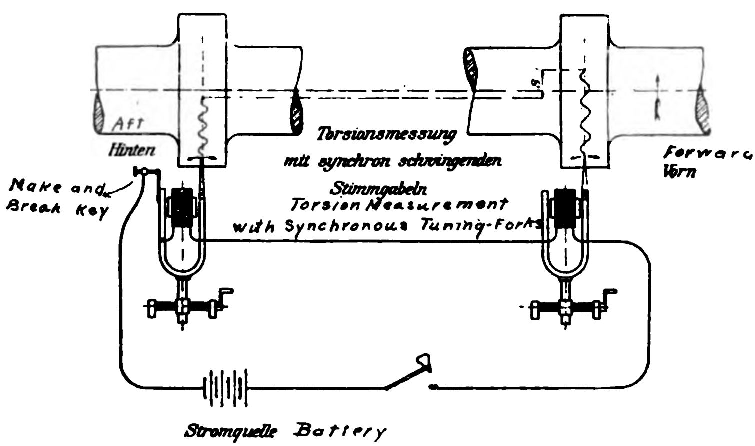


Fig. 17.

using one make-and-break for both. This arrangement is shown in Fig. 17. If the forks have the same rates of vibration they will, when actuated by the same current, strike out their arcs simultaneously. If the forks strike these arcs on the paper with a pencil, their intersection with the wave lines will give us a means of measuring the torsion.

Before the experiment the zero marks must be put on the drums for some definite crank position, such as "beginning of stroke."

Since the shaft always lies with an appreciable amount of torsion in the line bearings, stern bearings and stern-tube stuffing box, the zero mark is obtained by turning once forward and once backward, making a mark for each, and taking the point midway between these two for the real zero.

The amplitude of the vibrations of these forks was very small and caused the waves to be very long and flat, so I armed the prongs with extensions which actuated a small lever. The amplitude was thus multiplied fourfold.

There will not be true synchronism unless the frequencies of all the prongs are exactly alike. The frequencies may be adjusted by careful filing or by attaching very small weights.

I cannot enter into results of these interesting oscillation observations, but indicate the way (Fig. 18) to make syn-

Arrangement to test for exact Synchronism

Anordnung zur Prüfung des genauen Synchronismus.

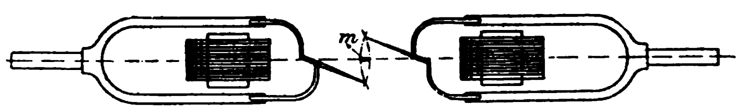


Fig. 18.

chronism of forks whose frequency exceeds 100 per second visible to the naked eye. The forks are placed opposite to each other, so that in a position of rest one pointer covers the other. When the circuit is closed one pointer swings to the right and the other to the left, and in the case of exact synchronism always pass each other at *m*. The crossing point *m* is easily observed in the veil formed, and with the least

change of phase it shifts to one side or the other. This method has even been used to demonstrate the phase variations of alternating currents.

Another method of testing for synchronism consists in making each pointer mark on a rotating drum, and regulating the distances of the marks from the starting point.

I had first resolved upon this tuning-fork method in connection with the torsionmeter diagram, but for lack of time I adopted, at the last moment, the use of electric flashes, which required less preparation and which I shall describe later.

METHOD WITH TWO SYNCHRONOUS MOTORS.

The difficult synchronizing of the two recording pencils is avoided by using small rotating synchronous motors. In connection with their principle it will be remembered that with electrically-controlled apparatus an exact motion like that of the hands of a clock is obtained by the use of certain electric arrangements like that of a revolving field.

If, with a revolving field, we did not restrict the pointer to one turn, but permitted it to make several, there would have to be an equal number of marks on the drum. If we put two motors in series they would revolve synchronously, coupled, so to speak, by electric force, like the hands of two electric clocks.

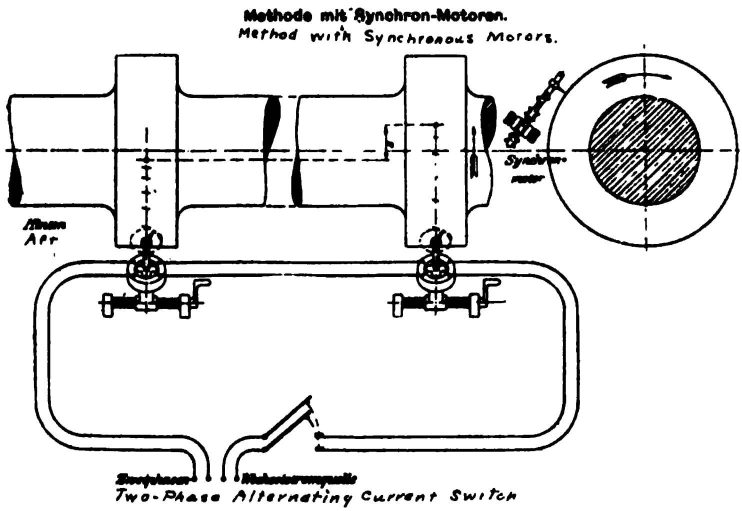
In Fig. 19 it is seen that with every rotation there is a mark made on the lamp-blacked paper; the circumferential line may thereby be cut at a good angle.

An alternating current is not always available, so the direct current of a ship's lighting system may be converted in a transformer.

A much simpler principle is embodied in the

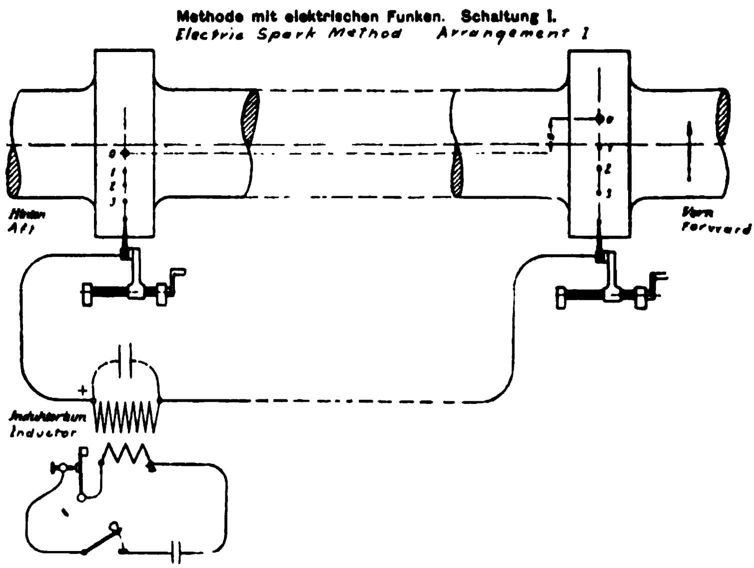
METHOD OF ELECTRIC FLASHES,

which I have applied with success to the torsion meter. If we insert two spark gaps in the secondary of a spark coil instead of the customary one, the spark will jump across both



Methode mit Synchron-Motoren.
Method with Synchronous Motors.
Aft
Two-Phase Alternating Current Switch

Fig. 19.



Methode mit elektrischen Funken. Schaltung I.
Electric Spark Method Arrangement I
Aft
Vorn
Forward
+
Inductor

Fig. 20.

gaps at exactly the same instant and perforate a sheet of paper inserted in each opening.

This simple idea is shown in Fig. 20. The two poles of the secondary of an induction coil consist of two metallic points which are distant about 1 mm. from their respective drums or collars. The shaft then forms part of the circuit. As soon as the interrupter begins to work sparks jump across the two gaps to the rotating paper-covered collars and make distinct holes in the paper.

A variation of this method is shown in Fig. 21. There are here two induction coils, whose primaries are in series with each other and are actuated by the same interrupter. The

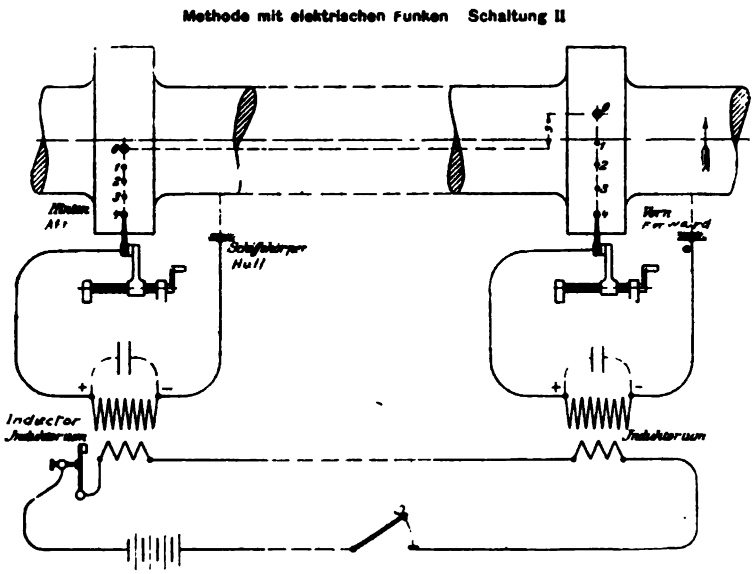


Fig. 21.

positive poles of the secondary are connected with the metal points, and the negative poles to the shaft (and thereby the ship's structure). As before, when the interrupter operates, sparks jump across their respective gaps and perforate the paper on the collars.

Both arrangements have been successful in practice. The second has the advantage of having no wiring between the two

secondaries and being more easily isolated in the shaft alley than the first. On the other hand, two inductors and a strong battery are necessary. To obtain very distinct holes Leyden jars are inserted in the secondary circuit.

To locate the holes easily it is advantageous to use paper blackened by a turpentine flame, for the spark blows the black deposit away for a radius of about 1 mm., so that the points appear to be white.

To keep the metal points the same distance away from the paper a polished hard-rubber ring was mounted on the collar, and this made a white line along the row of holes.

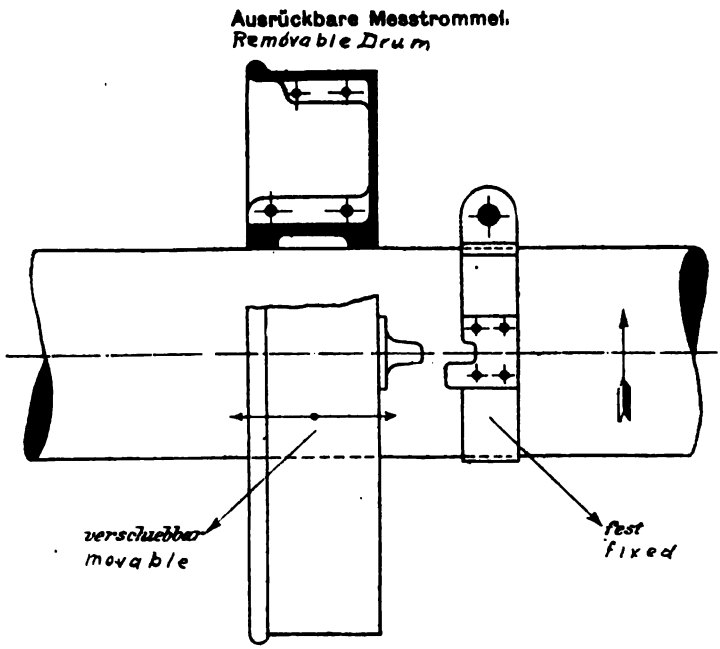


Fig. 22.

In order to avoid the trouble and loss of time necessary in locating the zero marks for each diagram, and to avoid stopping the engine, special removable measuring drums, 764 mm. diameter = 2,400 mm. circumference, were used, Fig. 22. The adjacent collar attached to the shaft has a wedge-shaped opening in which a, the wedge on the movable drum may be

inserted. By this arrangement an exact coupling between drum and shaft is obtained, and there is the advantage of being able to attach any desired number of papers to the drum while the shaft is revolving.

It is assumed that the zero mark has been permanently and accurately marked on the drum as well as on the diagrams.

There is the further advantage of being able to select a drum large enough in diameter to give a much larger arc of torsion than could be obtained on a shaft collar.

FRAHM'S METHOD.

This utilizes in an entirely different way the electric current for a simultaneous recording of the marks, and is decribed in the "Zeitschrift der Vereinigten Deutscher Ingeneure" (Proceedings of the Association of German Engineers), Nos. 22 and 24, 1902.

The chemical action of an electric current is used to make permanent marks on specially prepared zinc bands with the aid of two fixed platinum points.

For further details the reader is referred to the above publications. The use of the shaft collar increases the arc of torsion, but, for every new diagram, the engine must be stopped and the zero point be located anew.

Mr. Frahm experienced some difficulty in finding the corresponding rows and marks, for the contact points did not strike the shaft collar simultaneously, and, therefore, the first rows of marks on both collars did not correspond. I, therefore, made the suggestion that the circuit be not closed until the metal points had been moved axially along for one or two revolutions. The first marks on the two drums will then obviously correspond. This objection against the reliability of the result is thus easily removed.

PROFESSOR DENTON'S METHOD.

In regard to this I can only state that the efficiency of turbines is obtained from the torsion of the shaft by means of electricity. Further details I owe to private information.

Results are obtained from these diagrams by measuring the distances of the marks from their respective zero points and subtracting them from each other. These differences—that is, the torsion arcs—are transferred to another diagram.

If the marks are made at equal time intervals we can calculate the speed of the shaft at any instant from the distance between successive marks. For torsion measurements the length of time interval is immaterial.

The principal objection to the described methods consists in their intricacy. The operation of two measuring instruments with two diagrams, the easy failure of either, the necessity of having three observers, and, finally, the tedious calculations, are a great strain on the patience of the experimenters.

METHOD WITH ONE DRUM.

This is a distinct advance in torsion measurements. The differencing of the torsion marks is taken care of by the apparatus itself. The after drum carries a metal band having

Torsion Measurement with 1 Drum.
Torsionsmessung mit 1 Messtrommel.

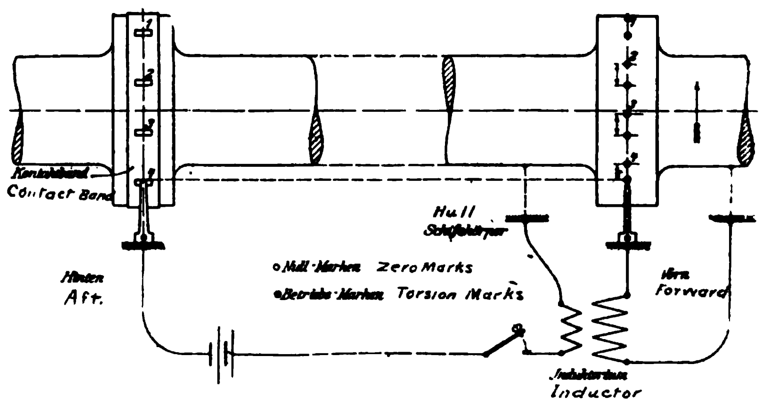


Fig. 23.

a series of insulated spots, 1, 2, 3, 4 (Fig. 23). As the shaft revolves these spots pass under a brush and thereby break the circuit of the primary of an induction coil located near the forward measuring drum.

At the instant of each break there is produced in the secondary circuit a spark which perforates the paper on the forward drum. Then, by slow rocking back and forth of the shaft, the zero marks corresponding to the breaks are located.

If the shaft were to turn without torsion the spark points would all fall exactly on the mean zero points. Actually, however, the working marks fall some distance from their zero marks, and the torsion for each position at no load can then be measured.

It is to be noticed in practice that in going ahead one edge of the break is the effective one, and in going astern the

Anordnung einer besonderen Rückwärtsbürste.
Arrangement with Special Backing Brush

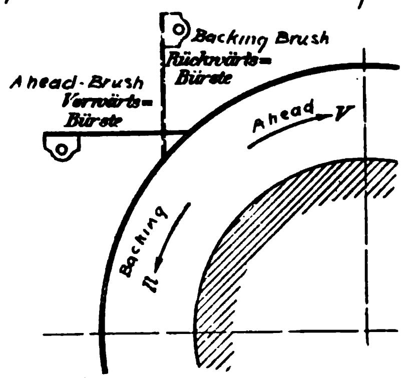


Fig. 24.

opposite edge is effective, and, therefore, the ahead and astern zero marks do not coincide.

This difficulty is avoided by a two-brush arrangement, shown in Fig. 24, by means of which the ahead break is made at the same instant as the going-astern break, and their zero marks would therefore coincide. The backing or astern brush is taken up after its zero point is made.

In the experiments there is nothing to do beyond placing the contact brush and the metal pole, whereby the circuits are completed, and the metal pole is then gradually moved axially along its drum. As before, the removable drum may

be used here to obtain several diagrams without stopping the engine.

An interesting variation (Fig. 25) of this method consists in making the torsions, and thereby the torques, visible by means of a luminous curve caused by the sparks. The spark gaps in this case are on the drum in a spiral curve around the drum.

The tangential distance (the ordinate) of the sparks from a given zero position is proportional to the torsion arc; the axial movement (the abscissa) of the spark is proportional to

Direct Representation of the Torsion Curve by a Chain of Sparks
Direkte Darstellung der Verdrehungskurve durch eine leuchtende Funkenkette.

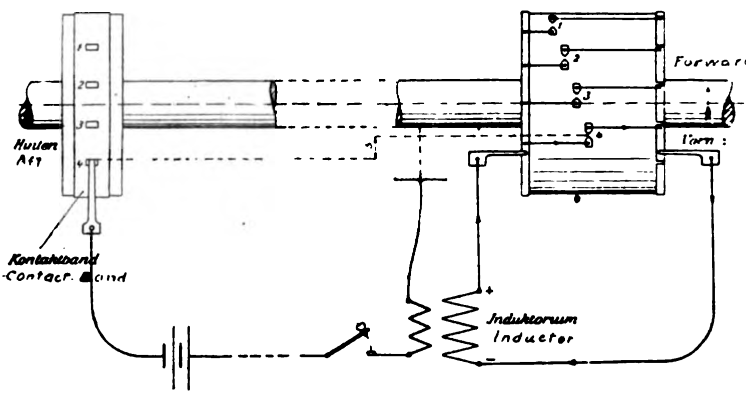


Fig. 25.

the angular twist of the forward drum. With the aid of a pane of glass the curve for each revolution can be traced on tracing paper.

I would have applied this method to a model if I had had it in time.

The use of the diagram by this last method and modification is very simple, and the torsion arc at any instant can be taken off directly.

I had intended to determine the torsion in this way on board a cruiser, but had, unfortunately, to satisfy myself with the calculation that the available shaft length at 10,000 H.P.

would give only 3 mm. torsion at the circumference of the collar. To calculate the torque from so small a measure is practically impossible; one needs at least a torsion of 15 mm., that is, a shaft length of 20 to 30 meters. This is a serious difficulty, one that shuts out from warship use all the foregoing methods.

There are also other difficulties. A shaft of that length consists of 4 or 5 sections and has 8 or 10 collars, concerning whose torsional behavior practically nothing is known.

Since the distribution of load along the shaft is not uniform, in the case of a long shaft we are reduced to inaccurate calculations, whose errors extend to all further results.

Moreover, such a shaft would necessitate 8 to 10 bearings, each one of which absorbs some energy.

In order to have unobjectionable results, the length of shafting used must be free from flanges and bearings. We are therefore restricted to the length of shaft between any two bearings, often only 2 or 3 meters in length.

In the case of a merchant steamer, a 2-meter shaft length, at a radius of 400 mm. gave a torsion of only 1.8 mm., with 2,000 H.P. and 76 revolutions.

Since in all similar cases the torsion arc is only 1 to 3 mm. (in the case of small warships it may be 6 to 8 mm.), it follows that improvement must follow not on the observer's end, but on the machine's end of the measuring. Otherwise, the errors of observation would often exceed the actual measurement.

The principle of torsion measurement with short shaft lengths (1 to 2 meters) is shown in Fig. 26. There is clamped to the shaft a sleeve which is free at its other end and carries a disc at the free end. Opposite this disc is another made directly fast to the shaft. If torque is applied to the shaft the two discs have a motion relative to each other in proportion to the torsion angle. The torque might then be measured from the divergence of two corresponding points on the discs. Every change in torque would cause a corresponding movement of the corresponding points.

For practical measurements arcs of 1 to 3 mm., and even of

Prinzip der Torsionsmessung auf kurze Messlänge.
Principle of Torsion Measurement with Short Shaft Lengths

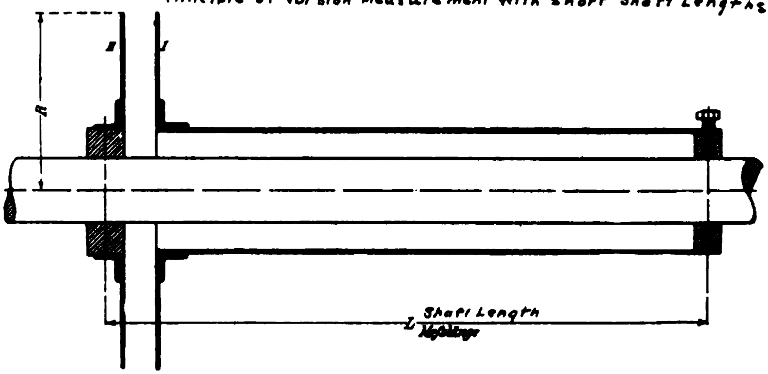


Fig. 26.

6 to 8 mm., are much too small; we may, however, let the measuring gear magnify or multiply this.

The principle of multiplying or magnifying is applied to all the following methods.

There is another point to be observed. The shaft and discs rotate while working while the observer is stationary; we must make the relative movement of the discs quantitatively visible.

In an extremely simple way both points are solved in the

Enlargement and Transfer of the Torsion Arc by Diagonal Slots.
Vergrösserung und Uebertragung des Verdrehungsbogens durch schräge Schlitze.

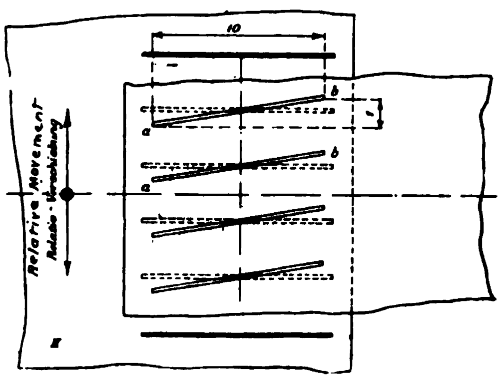


Fig. 27.

enlargement and transfer of the torsion arc by means of diagonal slits.

A white-painted sheet of tin, *I*, has at equal intervals slightly diagonal, parallel slits, *ab*. The tin sheet *II* has at the same intervals horizontal black lines (Fig. 27). If *I* is superposed on *II* a very small portion of each black line is visible through the slits. Every relative movement of the plates in the direction of the arrows causes a distinct movement of the visible black portion between *a* and *b*. If the angularity of the slits is 1 to 10 the movement of the black spots is ten times that of the plates. The action is the same if the tin plates are rolled on two concentric cylinders whose axes are parallel to the black lines.

If the cylinders rotate, the black parts show as a dark ring on the white cylinder covering, and if we apply this arrangement to Fig. 26, every relative movement of the discs will cause an axial displacement of the black ring. The distance of this black ring from the zero point measures the torsion arc and the play back and forth of the ring represents the variation of the effective torque.

This arrangement is not to be surpassed for cheapness and simplicity, and permits the maximum and minimum torque to be read on a scale without any further complications. To obtain more points for the construction of a curve, a screen rotating with the shaft is used. This covers all but one or two of the slits, and may be moved through very small distances.

Enlargement and transfer of the torsion arc by means of a ray of light affords some finer measurements.

The discs *I* and *II* are perforated near their circumference by holes that are opposite each other when there is no load on the shaft (Fig. 28). At the height of the shaft axis a fixed arc light throws only a horizontal ray through two of the holes, thereby giving a zero point on a vertical scale. When the shaft is working the relative movement of the discs causes the line joining any two corresponding holes to be diagonal, and the light ray follows this line. Therefore the

spot of light on the scale is moved from the zero point through the distance S, which is proportional to the torsion s. As the shaft rotates, each pair of holes causes the spot of

Vergrösserung und Uebertragung des Verdrehungsbogens durch Lichtstrahl.
Enlargement and Transfer of the Torsion Arc by a Ray of Light

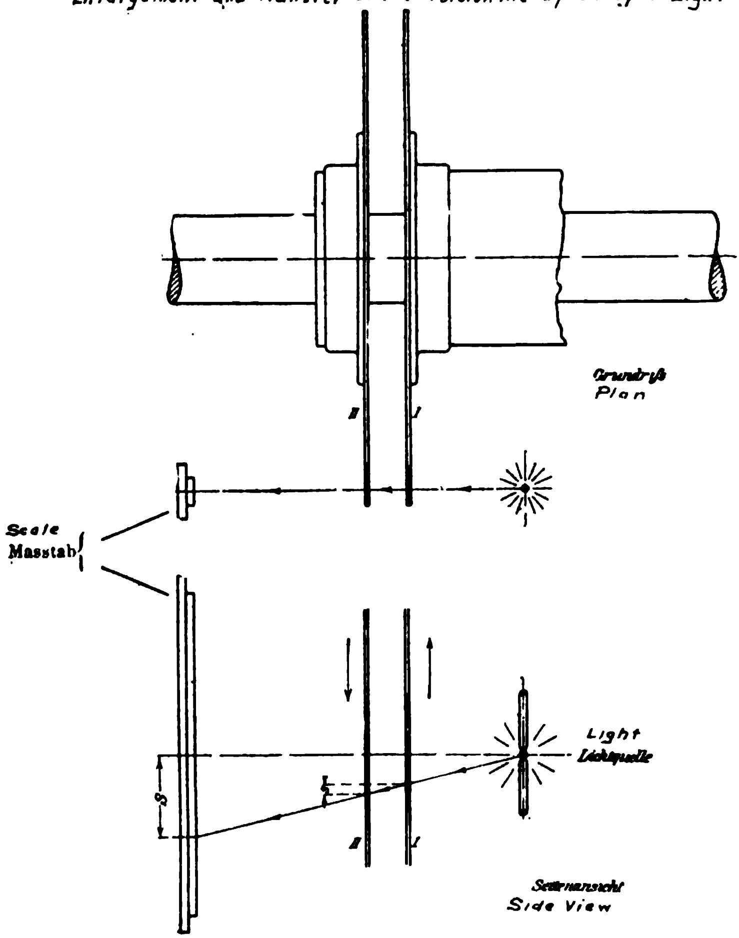


Fig. 28.

light to dance up and down the scale, picturing the variations in the torque.

In practice the light must be concentrated by means of

lenses and diaphragms. For the reading of single-torsion values a rotating screen may be used to advantage.

This system is better adapted to use in physical laboratories than in shaft alleys. Technical demands go further. From the technical standpoint it is not satisfactory to tediously get results from measurements obtained with difficulty. On trial trips, for example, the engineer wants to see immediate results in order to handle the engines to best advantage.

What is wanted is an arrangement that will give a diagram indicating the effective torque without the necessity for any mental effort.

THE TORSION METER.

This furnishes a solution of this comprehensive problem and is simple in principle. The small relative movements of the discs is multiplied by one or more levers in exactly the same way as is done with all steam indicators and manometers.

The principle difficulty lay in making the diagram obtained from the rotating discs easily accessible. To stop the engine is practically out of the question. A surprisingly simple solution is shown in Fig. 29. The levers *a*, *b*, *c*, *d*, *e*, *f*, *g*, are so

Pencil-Lever and Drum Arrangement of the Torsion Indicator
Schreibhebel- und Trommelanordnung des Torsionsindikators.

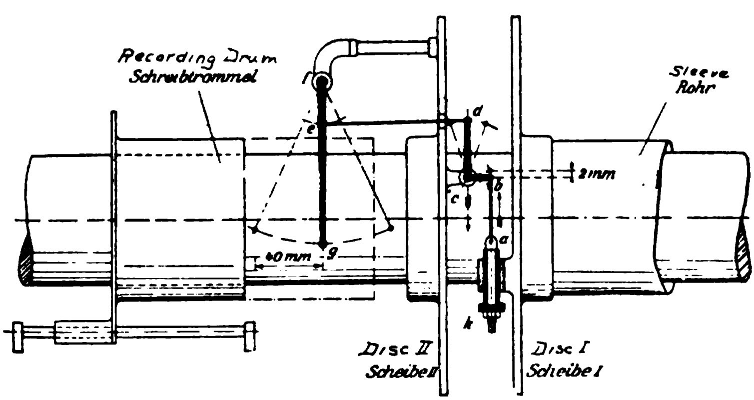


Fig. 29.

arranged that the pencil points to the center of the shaft and moves in the direction of the shaft's axis. Its displacement in this direction from the zero position therefore measures the instantaneous torsion of the shaft and thereby the torque.

A recording drum, concentric with the shaft, is arranged on a sliding drum so that it may be slid under the pencil when desired. In practice, the sleeve, discs and lever system all rotate; the pencil *g*, therefore, travels around the stationary recording drum and leaves on it the torsion curve.

When this drum is slid to one side, it is entirely free from the rotating part of the apparatus, and may be safely mounted with paper, or the finished diagram removed.

Of course, every diagram must carry a zero line similar to the atmospheric line of the indicator diagram. This is obtained from a second pencil rotating with the shaft, and permanently attached to it when the apparatus is first set up.

The ordinates of the diagram, that is, the axial movement of the pencil, are proportional to the torque at any instant; the abscissae, that is, the travel of the pencil around the drum, are proportional to the distance the ship has traveled; therefore, the area of the diagram, as is the case with the usual indicator diagram, represents the work done by the shaft. It follows that by means of a planimeter the mean torque and thereby the E.H.P. may be obtained. Figs. 30–34 show the constructive development of this idea for a shaft length of 2,200 mm., and a measuring radius, R, of 400 mm.

A cursory glance will betray the fact that the elegance of modern mechanical skill has been cast aside. Rather the details were constructed with reference to reliability and stability with the smallest weight and proper shape.

To discover and eliminate any one-sided working the relative disc movement was measured simultaneously at two diametrically opposite points, so that there were really two measuring systems, each regulating the other. All parts were made in duplicate, so that in the finished apparatus there are no corrections to be made.

The sleeve is centered on the shaft by means of four large

Torsionmeter Assembled

Torsionsindikator. Gesamtanordnung.

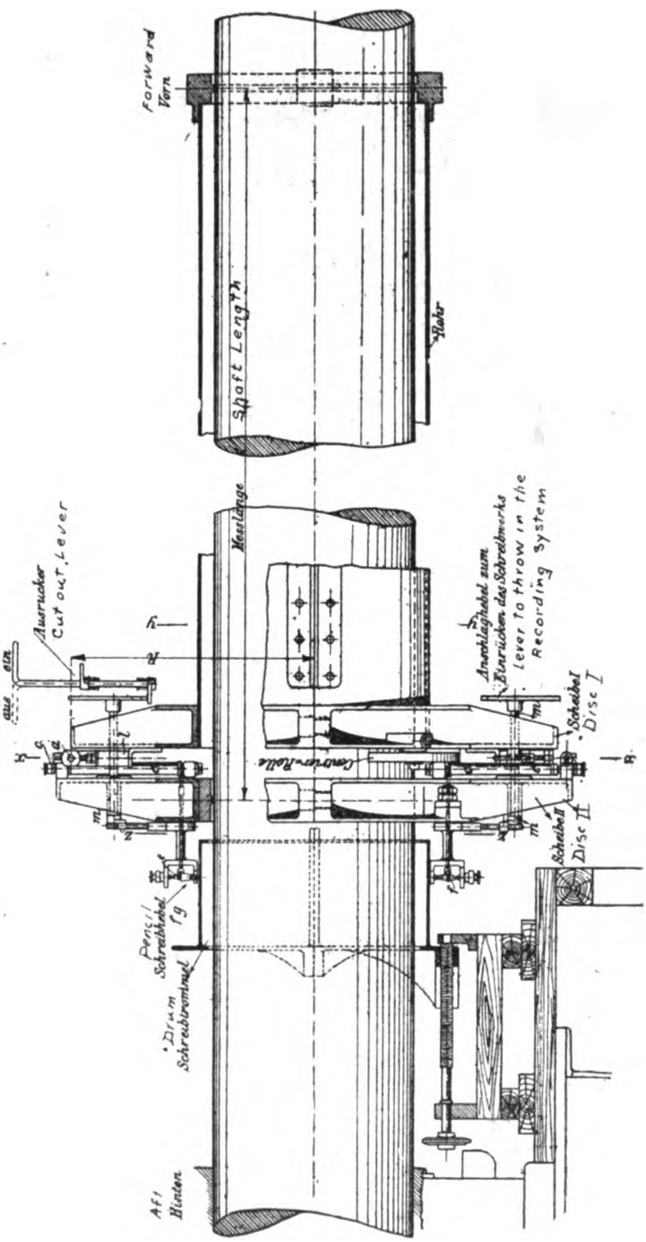


Fig. 30.

Plan
Oberansicht

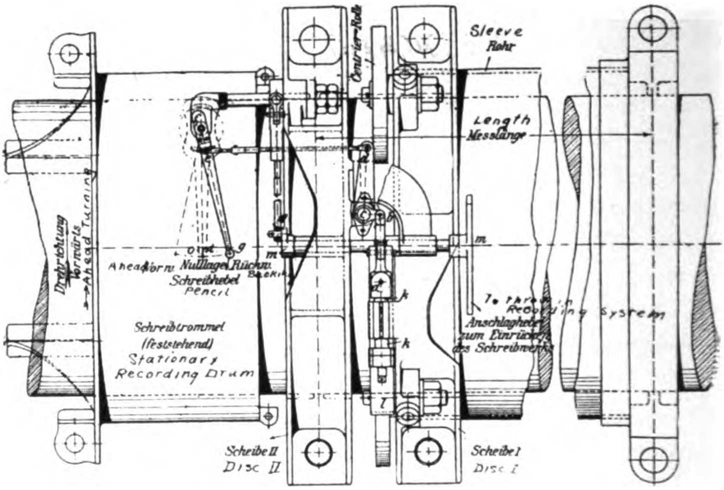


Fig. 31.

Cross Section
Querschnitt.

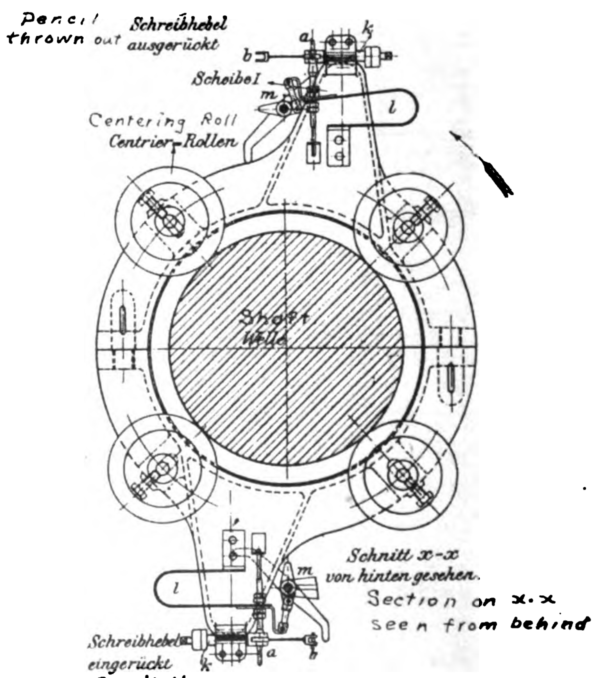


Fig. 32.

balls, adjustable by means of radial screws, and by means of these balls the friction is reduced to a minimum. In the arrangement for a 50-mm. shaft diameter roll sectors were used instead, because of irregularities on the shaft.

Following the principle of Fig. 29 the levers *a*, *b*, *c*, *d*, *e*, *f*, *g* (Fig. 31) multiply the relative disc movements 22 times, so that a mean torsion of 2 mm. would be represented by 44 mm. on the diagram. The greatest enemy to an exact measurement is in this case the lost motion in the joints. This was removed as much as possible by means of conical adjustable pins in all stationary parts, and carefully adjusted cylindrical pins for all moving parts. The residual lost motion was rendered harmless by means of a spring that kept each pin pressed in place.

In order not to complicate the apparatus at the start the slide guides of the steam indicator were not adopted, and no error in the measured efficiency resulted, as may be shown geometrically; for the rest, the always slightly distorted diagram may be transferred from polar to perpendicular coordinates, as shown in Fig. 43.

The pencil must be rather firmly pressed against the drum because of the centrifugal force. When the drum is slid into place the pencil would therefore break off if there were not some way of lifting up the pencil.

To accomplish this with the rotating apparatus there is a small spindle *M*, with a crank on one end and a lever on the other, which works on the principle of the star of a ratchet brace. This permits the pillars with the fixed and the oscillating pencils to be turned through a small angle (Figs. 33, 34).

This arrangement also accomplishes another object, that of arresting the lever motion. It did not seem good to have the joints working during the whole run, although one does not consider that point in the case of steam gauges on receivers. The stopping is done by breaking the connections between the two discs. The conical pin *a* (Figs. 32 and 34) is drawn radially inward by a small crank and the spring *l*.

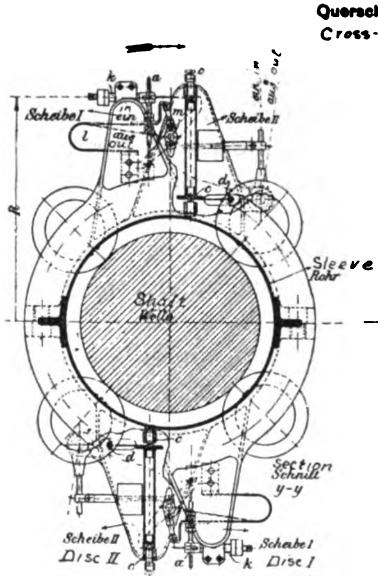


Fig. 33.

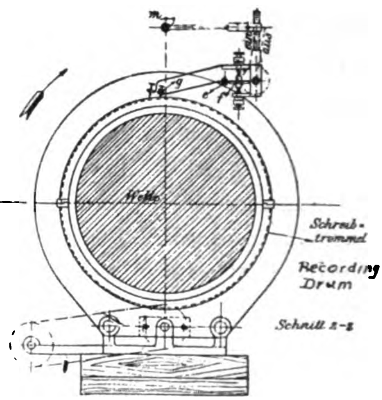


Fig. 34.

The motion of this withdrawing and stopping arrangement is effected by the striking of the right-hand lever (Fig. 30) on either the inner or the outer lugs of an adjustable arm attached to the ship's framing.

One of the chief difficulties in practice is the speed of the recording. With a drum diameter of 380 mm. and 80 r.p.m., the peripheral speed is 1.6 meters per second. With suitable recording arrangements it is possible to obtain good results with even double that speed, so that there is no objection to the use of this torsionmeter either in the largest steamships or in torpedo boats.

In this case a new recording-drum arrangement is used. The principle is shown in Fig. 35. The relative speed of the pencil and the drum is reduced by having the drum rotate with the pencil, but at a slower speed. This is accomplished by the spur gearing q, r, s, which drives the drum, but also permits its being stopped while the shaft is rotating. It is not worth while to go further into the constructive de-

Rotating Drum with Decreased Recording Speed
Rotierende Schreibtrommel mit verminderter Schreibgeschwindigkeit.

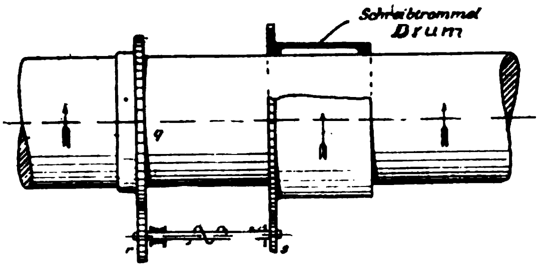


Fig. 35.

velopment of this idea, which has been worked out in an entirely different way by means of small recording drums.

It is hardly necessary to state that the principal idea of the apparatus permitted an endless variety of details, most of which were rejected. Among these was a hydraulic increase in the torsion, and an electrically controlled movement of the pointer at any desired part of the ship.

Possible errors resulting from torsional variations of the sleeve, due to varying torque on the shaft, were avoided, as in Fig. 36. The torsional vibrations of such short sleeves is enormously high.

Arrangement for prevention of Sleeve-vibrations
Anordnung zur Verhütung von Torsionsschwingungen des Rohres.

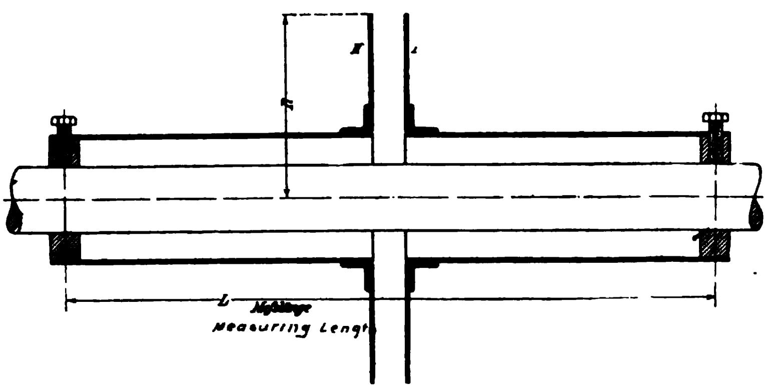


Fig. 36.

If there is little shaft room the recording drum may be placed on the sleeve (Fig. 39), as was done with the model for 50 to 140-mm. shaft diameter.

With steam turbines only one measurement is necessary, owing to the absence of pistons and cylinders; and because of the constant torque the inertia of the levers may be neglected, so that the axial movement of the levers, that is, the torque, may be read directly from a fixed scale, as in Fig. 37.

Torsion Indicator for Turbine Ships with Direct Reading of Torque
Torsionsindikator für Turbinenschiffe mit direkter Ablesung des Drehmoments.

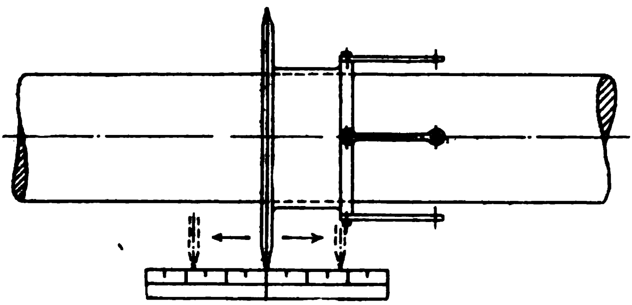


Fig. 37.

The torsionmeter may be attached to smaller shafts by means of an interchangeable case or by means of four set screws.

The apparatus may be made smaller if we separate it from the shaft (Fig. 38) and drive it by means of gearing. In this

Torsion-Indicator with Spur Gearing
Torsionsindikator mit Zahnradantrieb.

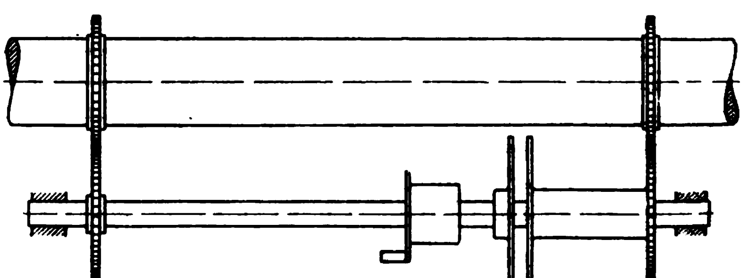


Fig. 38.

case there is a loss due to lost motion and the increase in parts.

The sensitiveness may be varied by changing the radius of measurement or altering the lever system.

HANDLING OF THE INDICATOR.

Before discussing the experiments the method of handling the apparatus is shown. First the position of the pencil for the torque-free shaft is ascertained through slow back and forward rocking of the shaft. Zero marks, some millimeters apart, are thus obtained. Assuming the propeller to be accurately balanced, the true zero line is the mean of these, and there the stationary pencil is fixed.

Then is obtained the position of the pencil on the drum corresponding to the beginning of the piston stroke, by rocking the shaft as before, and taking the mean.

The actual taking of the diagram consists, besides sliding the drum back and forth, in nothing more than turning the cut-out lever (Fig. 30) to the "in" and "out" positions. The entire manipulation was done by a locksmith, given the directions but once.

THE EXPERIMENTS.

The first experiments were made with a rather small model on the power shaft of a punching machine.

Naturally the fairly uniform torque was hardly interesting, because comparisons cannot be made with only one calculation. The starting period only was interesting, and began with very jerky vibrations.

Further tests were made on a shaft of 50 mm. diameter mounted on the centers of a lathe (Fig. 39), and weighted with a belt pulley and heavy lead weights.

On turning the shaft the weight caused a sinuform turning moment, for the pointer moved sinuformly. The curve to be expected was a sine curve symmetric to the zero line.

Fig. 40 shows copies of the curves obtained under four different conditions. They show the course of the torque and the load.

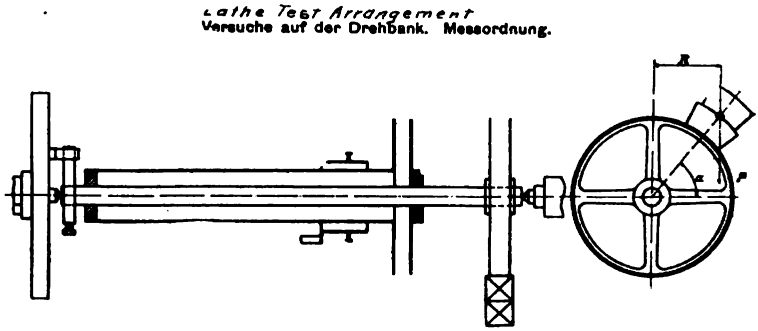


Fig. 39.

The position of the zero line is not exactly symmetrical to the sine curve. This is due to a constant torque necessary to overcome the friction of the after lathe center. By using a planimeter on the diagram the corresponding work could easily be found. By tightening up the lathe center the load could be increased at will.

Even with the low speed of 27 revolutions per minute there are distinct variations from the sine curve near the zero

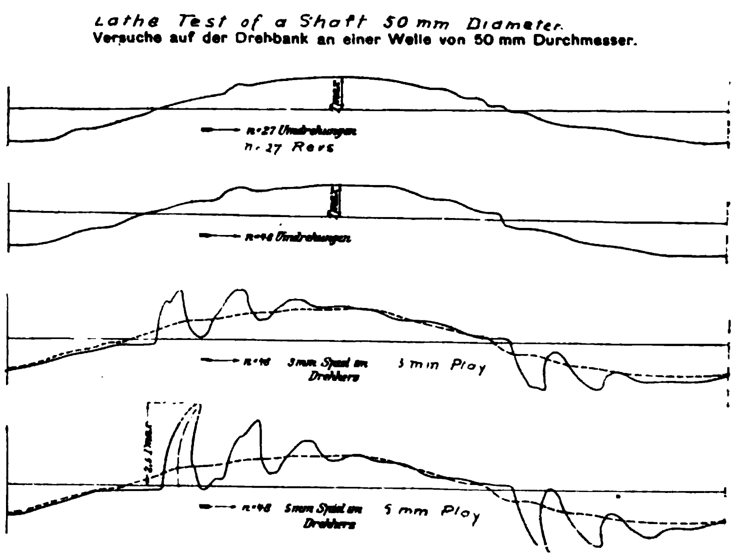


Fig. 40.

line, that is, at the shifting of the load. The cause could be heard in the loud rattling of the spur teeth. The play of the teeth naturally causes vibrations which appear as torsional vibrations of the shaft. This increases with the revolutions, as the second diagram shows.

For special reasons the tests were limited to the cases where there was 3 and 5 mm. play at the dog. The knocking that resulted called forth torsional vibrations which demanded double or three times the maximum static torque. Unfortunately, therefore, the first amplitude after each knock could not be accurately known, since the torsion indicator was not built for such large variations.

The resultant curve is not exactly sinuform because of great torsional vibrations, the more so because of the curved coordinates and of the fact that the abscissae represent angles of advance and not intervals of time. In addition, there are waves on the curve of a still higher period which may be attributed to local vibrations.

From the results far-reaching inferences may be made, for example, in regard to the extent of bending vibrations caused by jolting, and their demands.

If a play of 3 to 5 mm. in an elastic shaft of 50 mm. diameter and 3 m. length requires a power 2 to 3 times the normal driving power, what may we not expect of tail shafts which often have as much as 10 mm. play in the stern bearings and are jolted around by a poorly-balanced propeller?

If we consider that the lateral or bending vibrations are much harder to take up than the twisting vibrations just described, it is easy to see that in the long run the shaft cannot withstand the enormous overstrains.

After the information obtained from the first tests a large indicator was made for a shaft 320 mm. diameter. It was mounted on the three-crank shaft of a merchant steamer of 2,200 I.H.P. and about 75 revolutions.

In spite of the great difference in the powers measured (that of 50 mm. and a 320 mm. shaft), and the fact that the indicator was assembled at the last minute, all parts of it func-

Path of Torsion-Indicator Curves at Different Revolutions
Verlauf der mit dem Torsionsindikator genommenen Diagramme bei verschiedenen Tourenzahlen.

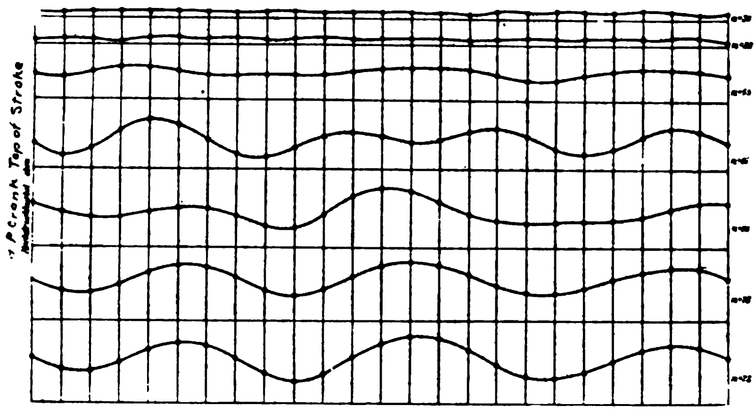


Fig. 41.

tioned satisfactorily from the start. Minor accidents at the beginning, such as the flying off of a pencil point or the jarring of the recording lever, do not at all subtract from its complete success.

For example, a series of the original diagrams are shown in Figs. 41–42, taken at different revolutions and referred to curved ordinates. The graphic method of transferring them to rectangular ordinates is shown in Fig. 43. By using the

Torsion Indicator Curves at Different Revolutions
Verlauf der mit dem Torsionsindikator genommenen Diagramme bei verschiedenen Tourenzahlen.

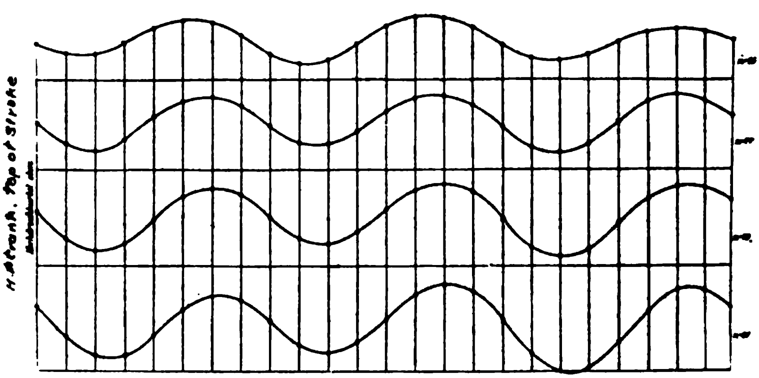


Fig. 42.

planimeter on the original diagrams, the mean torque and therefore the E.H.P. of the engine was obtained. The modulus of elasticity used was that ascertained by Frahm, 830,000 Kg/cm^2 (11,801,936 pounds per square inch).

Transfer of the Original Curve from Curved to Rectangular Ordinates.
Umzeichnung der Originaldiagramme von Bogenordinaten auf rechtwinklige Ordinaten.
Curve I Originaldiagramm. Curve II umgezeichnet.

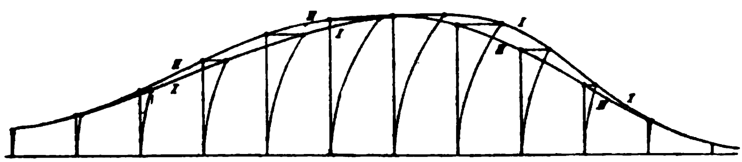


Fig. 43.

Since two sets of levers and pencils were set opposite each other, each recorded half the diagram and gave thus a good measure of their accuracy.

The horsepowers obtained vary, for the most part less than 1 per cent., and only at the lower powers as much as 2 per cent.

The developed curve of E.H.P. revolutions is shown in Fig. 44. The poor position of many of the points may be traced

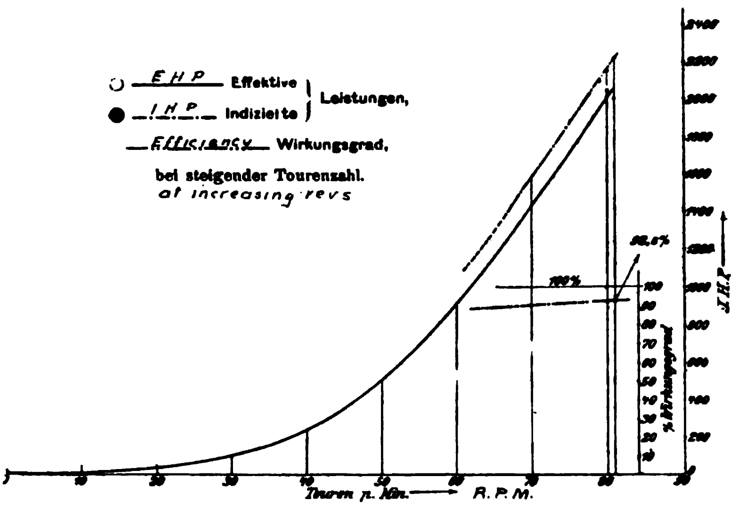


Fig. 44.

back to inaccurate counting of the revolutions with the help of a watch, something which is possible even on the largest merchant steamers for lack of a proper tachometer, as the revolutions are not constant. This disadvantage is especially evident in the case of steam indicator diagrams, because the six diagrams of a three-cylinder engine cannot be taken at the same instant.

For this reason the mechanical efficiency may be reliably obtained only from a comparison of the I.H.P. curve with the E.H.P. curve, as has been done here; otherwise, an efficiency of over 100 per cent. would seem possible. The curve here given shows a slow falling of the efficiency with decreasing load, the maximum being over 92 per cent. with 2,200 I.H.P. or 2,000 E.H.P.

Of greatest interest would be a comparison of the torsion-indicator curves with their corresponding torque curves.

In Figs. 45 to 48 are shown, for four different revolutions, the torque due to steam pressures alone and then with the added moments of inertia of the reciprocating parts. For the higher revolutions 77 and 81 the moments of inertia were obtained corresponding to the variable experimental peripheral velocity, which varied but little from the mean peripheral velocity.

The lowest diagram shows in each case the theoretical torque curve alongside the effective torque curve. To exactly compare the latter with the former is not at all easy, for when the recording pencil passes the "beginning of stroke" point the H.P. piston is no longer at the dead point, but has advanced by the torsion angle. The author's ideas as to the proper combination of these curves may be found in the Appendix. This method was followed with all these diagrams.

Consider now Fig. 45 for 66 revolutions. The course of the resultant or theoretical torque diagrams is very favorable; it shows three fluctuations which vary but little from the mean torque. According to the usual views the ordinate of both curves ought to correspond very closely, but a cursory glance will show that this is seldom the case. The fluctuations

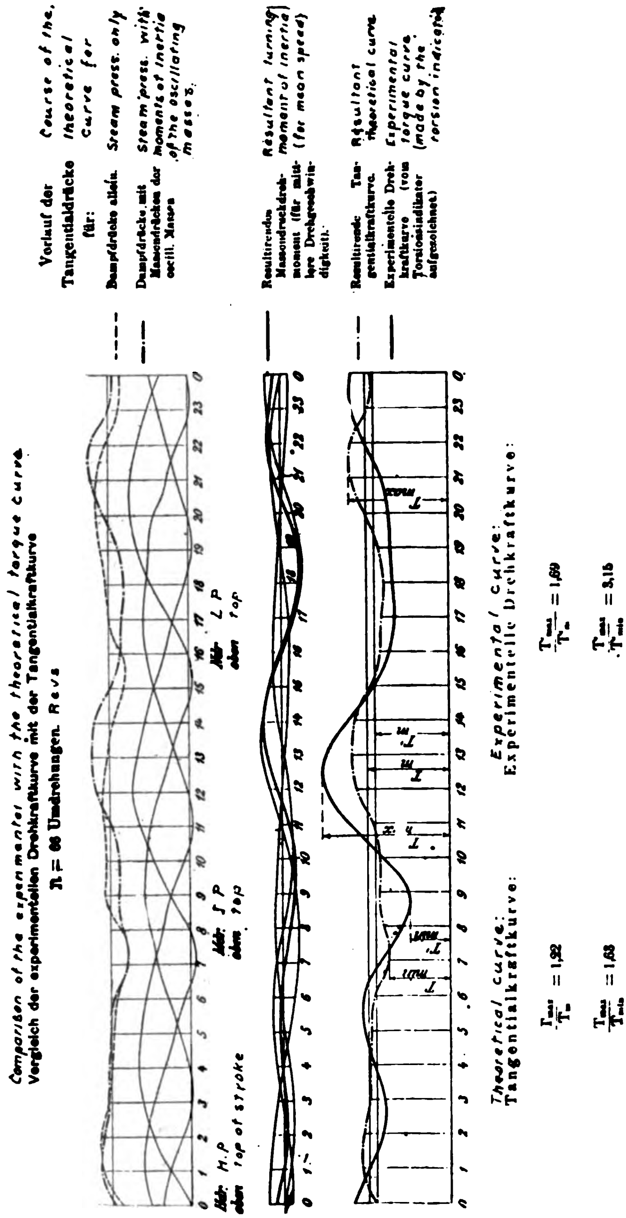

Comparison of the experimental with the theoretical torque curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve
n = 66 Umdrehungen. Revs
Hdr. H.P oben top of stroke
Mdr. J P oben top
Ndr. L P oben top
Vorlauf der Tangentialdrücke für:
Course of the theoretical curve for
Dampfdrücke allein.
Steam press. only
Dampfdrücke mit Massendrücken der oscill. Massen
Steam press. with moments of inertia of the oscillating masses.
Resultirendes Massendruckdrehmoment (für mittlere Drehgeschwindigkeit).
Résultant turning moment of inertia (for mean speed)
Resultirende Tangentialkraftkurve.
Résultant theoretical curve.
Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet)
Experimental torque curve (made by the torsion indicator)
Theoretical curve:
Tangentialkraftkurve:
$\frac{T_{max}}{T_m} = 1{,}22$
$\frac{T_{max}}{T_{min}} = 1{,}63$
Experimental curve:
Experimentelle Drehkraftkurve:
$\frac{T'_{max}}{T'_m} = 1{,}69$
$\frac{T'_{max}}{T'_{min}} = 3{,}15$

Fig. 45.

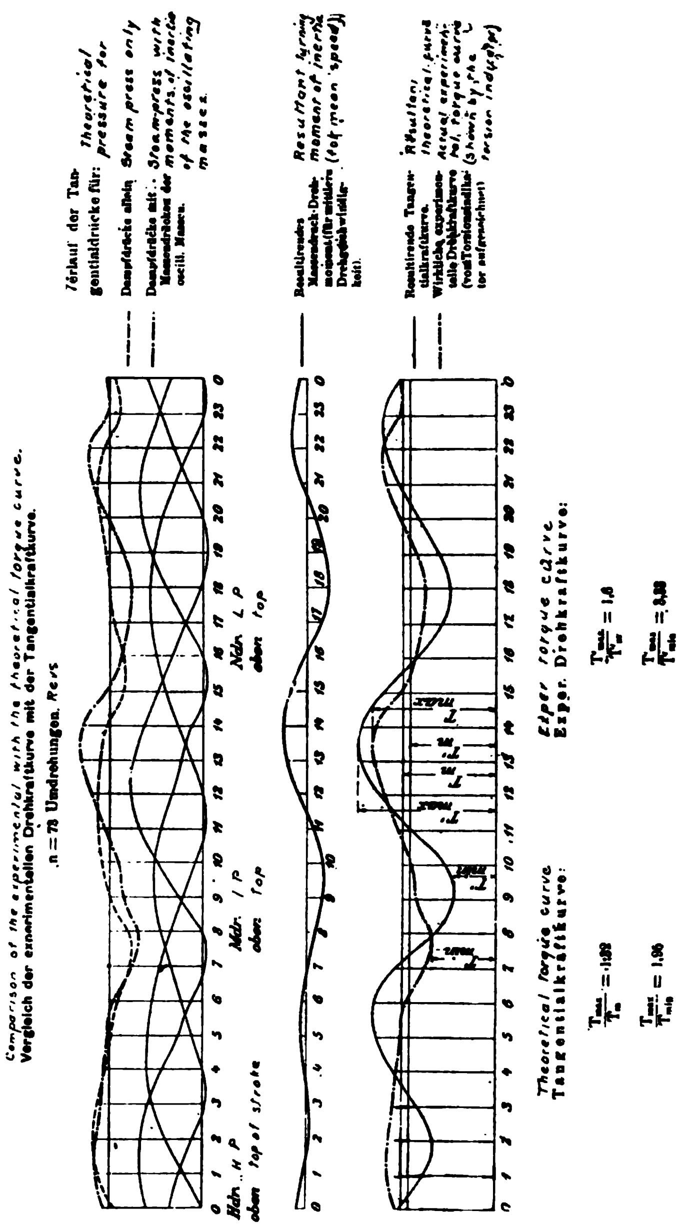

Comparison of the experimental with the theoretical torque curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve.
n = 78 Umdrehungen. Revs
Verlauf der Tangentialdrücke für: Theoretical pressure for
Dampfdrücke allein Steam press only
Dampfdrücke mit Massendrücken der oscill. Massen. Steam-press with moments of inertia of the oscillating masses.
Hdr. H P oben top of stroke
Mdr. I P oben top
Ndr. L P oben top
Resultirendes Massendruck-Drehmoment (für mittlere Drehgeschwindigkeit). Resultant turning moment of inertia (for mean speed)
Resultirende Tangentialkraftkurve. Resultant theoretical curve
Wirkliche experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet) Actual experimental torque curve (shown by the torsion indicator)
Theoretical torque curve
Tangentialkraftkurve:
$\frac{T_{max}}{T_m} = 1{,}32$
$\frac{T_{max}}{T_{min}} = 1{,}96$
Exper torque curve
Exper. Drehkraftkurve:
$\frac{T'_{max}}{T'_m} = 1{,}6$
$\frac{T'_{max}}{T'_{min}} = 3{,}33$

Fig. 46.

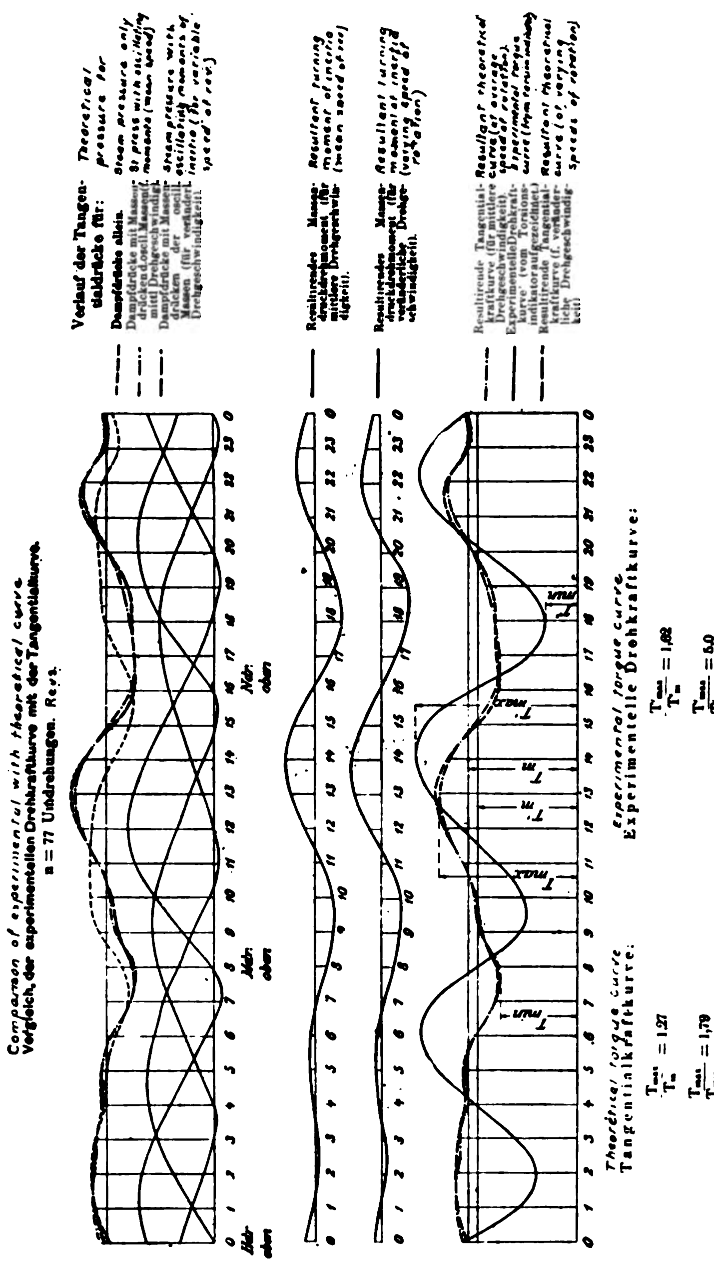

Comparison of experimental with theoretical curve
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkurve.
n = 77 Umdrehungen. Revs.
Verlauf der Tangentialdrücke für:
Theoretical pressure for
Dampfdrücke allein.
Steam pressure only
Dampfdrücke mit Massendrücken d. oscil. Massen (f. mittl. Drehgeschwindigk.)
St press with oscillating moments (mean speed)
Dampfdrücke mit Massendrücken der oscill. Massen (für veränderl. Drehgeschwindigkeit).
Steampressure with oscillating moments of inertia (for variable speed of rev.)
Resultirendes Massendruckdrehmoment (für mittlere Drehgeschwindigkeit).
Resultant turning moment of inertia (mean speed of rev.)
Resultirendes Massendruckdrehmoment (für veränderliche Drehgeschwindigkeit).
Resultant turning moment of inertia (varying speed of rotation)
Resultirende Tangentialkraftkurve (für mittlere Drehgeschwindigkeit).
Resultant theoretical curve (at average speed of rotation).
Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet.)
Experimental torque curve (from torsion indicator)
Resultirende Tangentialkraftkurve (f. veränderliche Drehgeschwindigkeit)
Resultant theoretical curve (at varying speeds of rotation)
Hdr. oben
Ndr. oben
Theoretical torque curve
Tangentialkraftkurve:
$\frac{T_{max}}{T_m} = 1,27$
$\frac{T_{max}}{T_{min}} = 1,79$
Experimental torque curve
Experimentelle Drehkraftkurve:
$\frac{T_{max}}{T_m} = 1,82$
$\frac{T_{max}}{T_{min}} = 5,0$

Fig. 47.

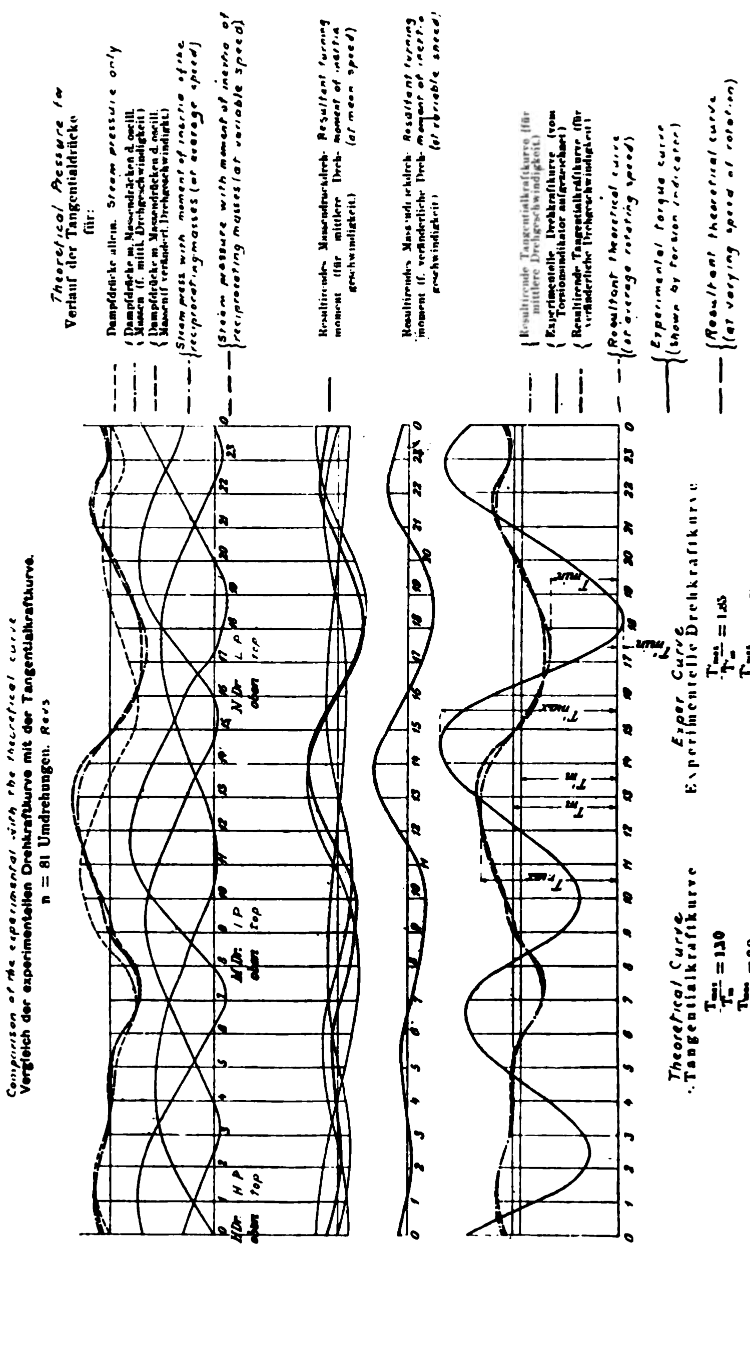

Comparison of the experimental with the theoretical curve
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve.
n = 81 Umdrehungen. Revs
H Dr. oben
H P top
M Dr. oben
I P top
N Dr. oben
L P top
Theoretical Pressure for
Verlauf der Tangentialdrücke für:
Dampfdrücke allein. Steam pressure only
Dampfdrücke m. Massendrücken d. oscill. Massen (f. mittl. Drehgeschwindigkeit)
Dampfdrücke m. Massendrücken d. oscill. Massen (f. veränderl. Drehgeschwindigkt.)
Steam press. with moment of inertia of the reciprocating masses (at average speed)
Steam pressure with moment of inertia of reciprocating masses (at variable speed)
Resultirendes Massendruckdrehmoment (für mittlere Drehgeschwindigkeit.) Resultant turning moment of inertia (at mean speed)
Resultirendes Massendruckdrehmoment (f. veränderliche Drehgeschwindigkeit) Resultant turning moment of inertia (at variable speed)
Resultirende Tangentialkraftkurve (für mittlere Drehgeschwindigkeit.)
Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet)
Resultirende Tangentialkraftkurve (für veränderliche Drehgeschwindigkeit)
Resultant theoretical curve (at average rotating speed)
Experimental torque curve (shown by torsion indicator)
Resultant theoretical curve (at varying speed of rotation)
T_{max}
T_m
T'_m
T'_{max}
T_{min}
T'_{min}
Theoretical Curve
Tangentialkraftkurve
$\frac{T_{max}}{T_m} = 1.30$
$\frac{T_{max}}{T_{min}} = 2.0$
Exper. Curve
Experimentelle Drehkraftkurve
$\frac{T'_{max}}{T'_m} = 1.85$
$\frac{T'_{max}}{T'_{min}} = -20.4$

Fig. 48.

of the effective torque are much greater than those of the theoretical torque and are partially opposed to it. The failure of the two to coincide is shown by the ratio:

$T\,max.$: $T\,min.$ = 1.22 × theoretical torque;

$T'\,max.$: $T'\,min.$ = 1.69 × effective torque.

Still more different are the values of $T\,max.$: $T\,min.$ = 1.63 to 3.15.

Fig. 46, for 73 revolutions, shows still greater difference. The lowest diagram shows the comparison. Following the two distinct fluctuations in the theoretical torque curve we might expect the same for the effective torque curve; but instead we have three distinct oscillations of the shaft, of well rounded form and without any minor variations. The fluctuations of the effective torque are here also much greater than might be expected, the ratios being $T\,max.$: $T\,min.$ = 1.32 and 1.6, and $T\,max.$: $T\,min.$ = 1.95 and 3.33.

Fig. 47, for 77 revolutions, shows still greater torsional oscillations, whose maximum amplitude is 1.6 times the mean torsion, and the theoretical diagram shows the ratio, $T\,max.$: $T\,min.$ = 1.27. Very noticeable is the difference of the opposite ratios of $T\,max.$: $T\,min.$, namely 5.0 and 1.79. Striking is the fact, shown in Fig. 46, for 73 revolutions, that the first oscillation of the effective torque is exactly opposite to the theoretical torque.

The comparison shown by Fig. 48, for 81 revolutions, is surprising. The two distinct oscillations of the theoretical torque ought to call forth only two fluctuations in the effective torque; but instead there appear three strong torsional oscillations per revolution, of which the last one actually assumes the nature of a negative turning moment, which would show that for an instant the advancing propeller dragged the combined shaft and moving engine weights around with it. It is, therefore, not to be wondered at that there appear the very different ratios of $T\,max.$: $T\,min.$ = 1.30 and 1.85 and $T\,max.$: $T\,min.$ = 2.0 and −29.4.

The only explanation for these differences occurring at various revolutions lies in the "Torsional Oscillations of the System."

A calculation of the oscillations of the shaft was made from this formula:

$$N = \frac{30}{\pi R}\sqrt{\frac{G\,\theta\,(M+m)}{L\cdot M\cdot m}},$$

where

R = crank radius,

G = modulus of elasticity in kg/cm^2,

θ = polar moment of inertia of the shaft's cross section in cm^4,

L = shaft length reduced to cm,

$$\left.\begin{array}{l} M = \text{propeller mass,} \\ m = \text{engine mass,} \end{array}\right\}\text{reduced to the crank circle.}$$

The result was 243 oscillations per minute, with 81 revolutions. 243 ÷ 81 = 3 torsional vibrations of the shaft exactly for one revolution, which would show 81 revolutions to be a "critical" speed. Indeed, the oscillations might be as high as 250, and the close agreement with the experiments is very good, as Figs. 42 and 48 show.

We may conclude further that with 61 revolutions we may expect 243 ÷ 61 = 4 oscillations. The experiment proves it, for with 63 revolutions we had four distinct oscillations.

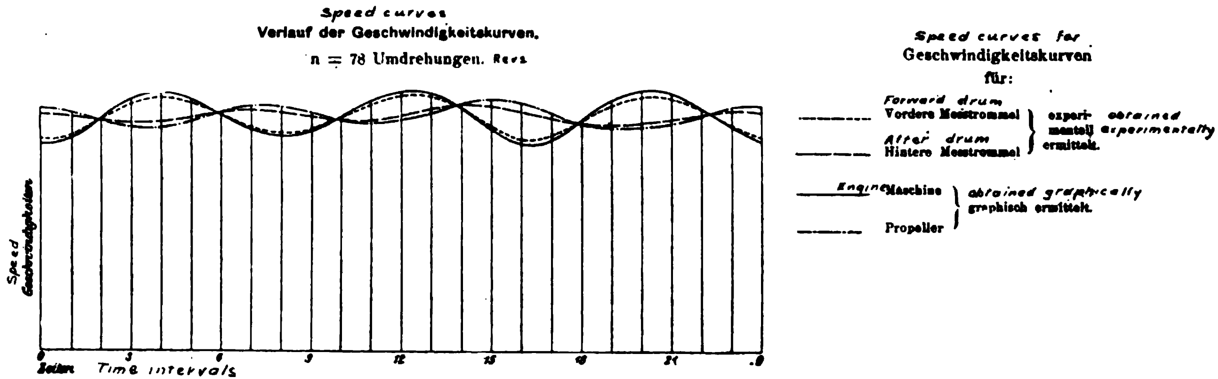
It may be further expected that with intermediate revolutions we will have a combination of the two frequencies of oscillation. Figs. 41 and 42 show this very plainly. The gradual change from a simple curve to the large oscillations at 81 revolutions demonstrates plainly the correctness of the calculated "critical" revolutions.

We might justly object to these assertions that the steam indicator would give exactly the same curves if its cylinder were made to oscillate with varying speed of the shaft.

Now, a rough calculation of the oscillations of the indicator will show that they are often several times as great in number as those of the shaft.

In order to remove the least objection and to test the accuracy of the indicator the torsion of a 27.5-meter shaft was measured by electric sparks with one or two inductors.

The torsion curves earlier obtained showed for correspond-



% of dissimilarity
Ungleichförmigkeitsgrade:

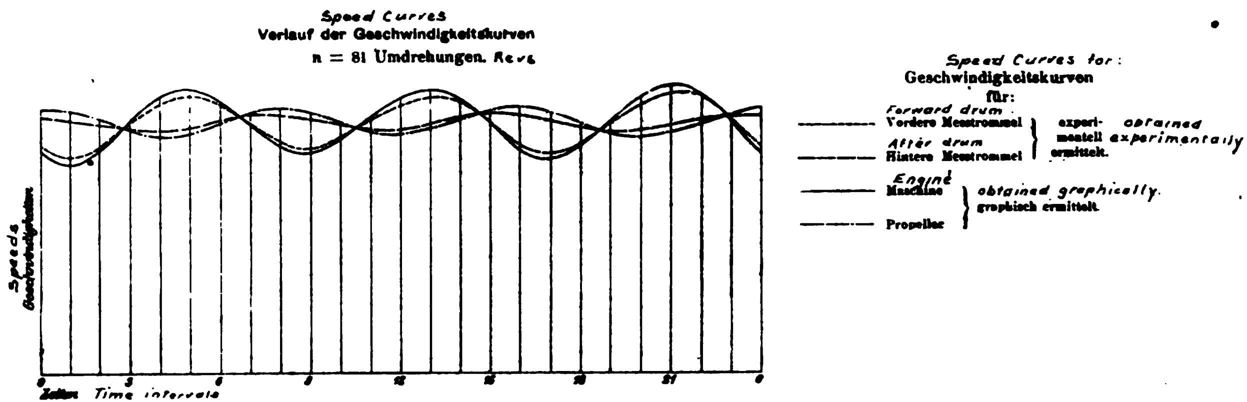
Engine Maschine: $\delta = \frac{v_{max} - v_{min}}{v_m} = \frac{1}{4,4} = 22,5\ \%$

Forward drum Vordere Messtrommel: $= \frac{1}{5,2} = 19,1\ \%$

After drum Hintere Messtrommel: $= \frac{1}{11,9} = 8,4\ \%$

Propeller Propeller: $= \frac{1}{8,5} = 11,8\ \%$.

Fig. 49.



% of dissimilarity:
Ungleichförmigkeitsgrade:

Engine Maschine: $\delta = \frac{V_{max} - V_{min}}{V_m} = \frac{1}{3,16} = 31,6\,\%$

Forward drum Vordere Messtrommel: $= \frac{1}{3,93} = 25,5\,\%$

After drum Hintere Messtrommel: $= \frac{1}{13,1} = 7,6\,\%$

Propeller Propeller: $= \frac{1}{8,7} = 11,5\,\%$

Fig. 50.

ing revolutions the same general form as the indicator diagram. To obtain a quantitative comparison of the results the difference *T max.* — *T min.* was obtained from different diagrams and the percentage of their variation calculated for both methods of measurement. Out of nine such calculations there was a mean difference of 1.2 per cent., which is regarded as sufficiently accurate for most physical measurements. Small inaccuracies in the vicinity of the zero line were eliminated by this difference method.

At the same time the varying speeds of the propeller and the engine were obtained from the spark diagrams of both drums by a method described in the appendix. Figs. 49 and 50 show the diagrams for 78 and 80 revolutions. The lack of uniformity varies for the different parts of the shaft, and at 80 revolutions that of the engine attains the high value of 31.6 per cent.

RESULTS AND CONCLUSIONS.

Having described the experiments, let us now briefly consider the results and the conclusions that may be drawn.

I. Most important is the fact that in the torsion indicator we have a technical apparatus which automatically, and without any mental effort, shows the effective torque in the form of a diagram, through the integration of which we obtain the E.H.P. Of great significance is the fact that shaft lengths of only 1 to 2 meters are necessary, while up to this time Frahm's method of torsion measurement required from 20 to 30 meters length.

Torsion measurement is therefore now made possible with ships, especially warships, whose shafting is short.

Another point is that all parts function perfectly, and the difficulties of the methods considered may be regarded as solved.

II. The close agreement of the tests made with entirely different methods shows the perfect reliability and accuracy of the diagrams. If we properly inform ourselves as to the moduli of elasticity of the shaft materials by means of experi-

mental shafts, we open a long perspective of practical as well as scientific measurements.

We are then in a position to ascertain from a single diagram the E.H.P. transmitted to the propeller, while, for example, 16 diagrams are necessary in the case of an eight-cylindered pair of steam engines.

This fact is of importance in connection with steam turbines, which we have no other way of indicating. The modification shown in Fig. 37 permits of direct reading of the torque from a scale.

The mechanical efficiencies of marine engines may also be simply obtained, and thus one of the most uncertain factors in the design of a new ship and of a propeller may be eliminated.

Obviously, it would be advantageous to science to profit by the easily-made torsion measurements, especially if, by the methods previously described, the variations of the revolution speeds be registered in the same drum. The actual speed of any point of the shaft may then be obtained from the single combined diagram. In the appendix it is further shown how to find the course of the frictional power and its overcoming in the engine. At the same time might be determined the energy absorbed by the ship's vibrations, a method for which may be found in the appendix.

III. It might be the case for special reasons that the first experiments with the apparatus would give quantitative results, but, of course, the test would have to be subordinated to the program of the trip.

In close agreement with the experiments of Frahm are the results here obtained, and they go to show the untenableness of the old theory that the actual torque is closely proportional to the theoretical torque.

Frahm's experiments were principally conducted either at the normal or the critical revolution speeds. The systematic tests described in the foregoing pages, and made at all usual speeds, prove the existence of torsional vibrations at speeds quite different from the critical speeds, and show the course

of the vibrations for a wide range of revolution speeds, something which is of the greatest importance in practical application.

The question is now, naturally, asked, how may these results be practically applied to the calculation of shaft dimensions?

It would be very wrong to wish to shake in the slightest the usual methods of design, namely, those of the various bureaus of standards, for they represent the decade's experience in all normal cases. The fact that the actual diagrams do not agree with the theoretical at all revolutions need not disturb us in the least, for this is taken in account, in a large measure, by the coefficients of the bureaus of standards or classifications. Very seldom have inexact coefficients been dangerous to the engineer, for it is the practice to copy former designs. Dangerous only are the authoritative view points which make insufficient allowance for abnormal cases, such as that where the normal revolution speed happens to be also the critical.

In order to avoid this it is possible, as Mr. Frahm has pointed out, to separate the critical from the normal speeds by changing the diameter of the shaft.

With reference to the practical execution of Mr. Frahm's proposal I should like to add the following:

1. If it is possible to calculate in advance the oscillations of the shaft only after having ascertained its elastic properties from exact model tests—that is, the torsion of the cranks, thrust shafts, flanges, couplings and shaft accessories—then primarily only a rough calculation is possible for the reduced shaft length, so that the calculation for the rate of oscillation could be only approximate.

2. Let it be emphasized that it is possible to change the calculated rate of oscillation in ships with long line shafting if the line shaft is weakened or strengthened, and that in this case it would be better not to change the diameter of the propeller and crank shafting.

It is of the greatest practical interest to ask how, in the

case of finished ships, existing dangerous torsional oscillations may be decreased in order to prevent a broken shaft.

A change in the rotating engine masses by the addition of a flywheel is seldom possible, and, even so, the rate of oscillations or vibration would not thereby be much affected, for in the formula used the engine masses appear under the radical in both the numerator and denominator.

In practice an easy way is to vary the work done in each cylinder by adjusting the link gear until the vibrations are reduced. The form of the theoretical diagram is thereby changed, as are therefore also its harmonic waves, which alone maintain the dangerous vibrations and determine their amplitude. We are indeed here restricted to rough trials; meanwhile, in my tests, I watched with safety for the possibility, even in a small degree, of dangerous vibrations.

A change in the harmonic waves set up and an eventual decrease in their amplitude may be obtained by changing the crank-angles, although that would be much more expensive than changing the positions of the links.

Finally, the dangerous vibrations may be overcome by changing the revolutions per minute. The tests described, for example, show a considerable decrease in the vibrations at the normal speed of 75 revolutions, as compared with those observed, the (in this case) exceptional speed of 81 revolutions. Of course, a change in the revolutions is often only slightly, or not at all, available, and then perhaps associated with many other difficulties.

IV. In conclusion, I wish to refer to the strains observed in the lathe experiments, to which we may compare the strains caused by jars and jolts in the propeller- and crank-shafts.

It now only remains for me to thank here the management of the Vulcan Engine Company, Stettin, through whose cooperation was made possible the development of the apparatus and the carrying out of the experiments. My thanks are also due to the graduate engineer, Mr. Jahn, who took part of the diagrams after my directions.

It would be highly desirable if my experiments were ex-

tended to warships, for the torsional vibrations measured at all higher revolutions seem to give, in most cases, a factor more or less useful in showing the needs of long marine-engine shafts.

APPENDIX.

COMBINATION OF THE EFFECTIVE TORQUE CURVE WITH THE THEORETICAL TORQUE CURVE.

To combine the effective torque curve given by the torsion indicator with the theoretical curve we must first reduce them to the same base.

Obviously, the ordinates of both must be reduced to the same pressure scale. The mark on the recording drum corresponding to top of H.P. stroke is good only for a torsionless shaft; during work the H.P. crank has passed the dead point by the torsion angle at the instant the indicator pencil marks the dead point. This holds good for every crank position.

Following is a method of combining the diagrams:

Both diagrams are divided into the same number, m, equal parts. Neither these division points on the two diagrams nor their ordinates correspond in point of time. One division represents an angle of 360 degrees $\div$ m. The torsion angle may easily be obtained in degrees from the length of the torsion arc. If, for example, for the nth division point of the diagram there is an angle of i degrees, then the corresponding point on the theoretical diagram is i degrees further on. In this way we obtain a new division of the theoretical diagram, of which every division point corresponds exactly with one on the torsion diagram. More difficult is the division of the theoretical diagram into equal time intervals, for which it is necessary to construct the integral curve of the speed diagram.

GRAPHIC METHOD OF DETERMINING THE SPEEDS OF THE PROPELLER AND THE ENGINE.

Available are the experimental speed curves of the after and the forward recording drums, best laid off with time intervals.

In Fig. 51 there are drawn four vertical parallel lines

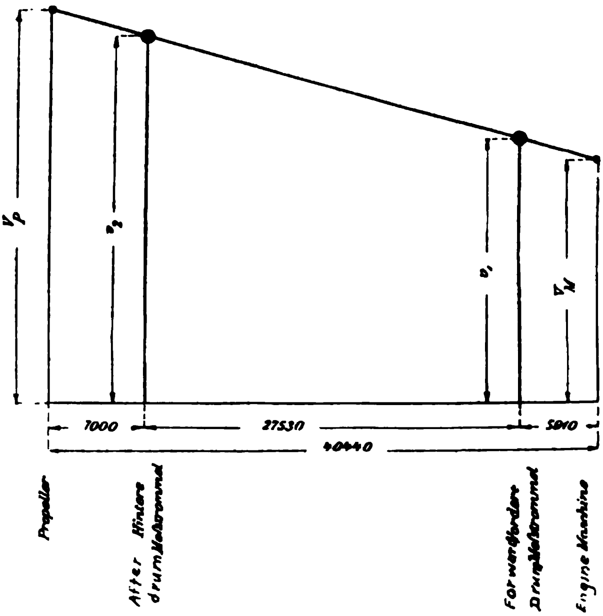


Fig. 51.

whose distances apart are proportional to their relative position on the shaft reduced to a uniform diameter. On any convenient scale are laid off V_1 and V_2, the simultaneous speeds measured on the two drums. A line produced through these two points cuts on the other two perpendiculars the desired speeds V_m and V_p.

The proof is as follows:

Four points which lie in a straight line on a torsionless shaft lie in a spiral line when the shaft is doing work, and this spiral line when unwound is an inclined straight line. This graphic construction shows the resultant unwinding at two different instants, the beginning and the end of the interval Δt. If in this time the two inside points have traveled the distances V_1 and V_2, so must the outer points have gone through the distances V_m and V_p. If $\Delta t = 1$, then V_1, V_2, V_m and V_p give the speed directly.

The base line need not be perpendicular to the parallel lines.

The peripheral speeds of any other shaft points may be found by drawing in the proper parallels. This construction gives the same results as those analytically obtained by Frahm.

DETERMINATION OF THE VARIABLE SPEED OF THE SHAFT BY THE TORSION INDICATOR.

This may be obtained experimentally as follows, from the normal torsion diagram.

The removable recording drum is insulated by a wooden or other base from the ship's hull and connected with the + pole of the secondary of an induction coil, whose — pole is connected with the shaft, and, thereby, with the torsion indicator and its pencils. If the primary circuit is broken at equal intervals the sparks in the secondary will pierce the diagram paper. Large distinct holes are obtained by placing a Leyden jar in parallel with the secondary circuit. In order to confine the sparks to one recording pencil the other pencils are insulated by hard-rubber strips.

The arc between two holes measured on a proper scale gives the speed at any instant for the corresponding part of the shaft.

The operation may be carried out successfully with the usual type of interrupter, but careful speed measurements require a mercury turbo-interrupter or some other motor type.

From the single combined diagram the speed for each shaft point may be found from the following considerations:

The measured speed of any one point of the shaft would be a measure for that of any other point if the shaft's torsion did not vary during the measuring interval.

In reality the speed of point No. 2 diverges from the measured absolute speed of point No. 1, while the relative speeds of points 1 and 2 come together, which is caused by a variation in the torsion arc.

The torsion indicator furnishes for the measured shaft length the multiplied torsion arc, and this is proportional to the torsion of any other shaft length.

The unknown relative speed may then be easily found by subtracting from each other the torsions at the beginning and

the end of the time interval and applying the proportional part of this difference to the shaft length between points 1 and 2.

DAMPING IN THE ENGINE.

The method is analogous to Frahm's determination of the coefficients of propeller resistance.

The action of the theoretical and of the effective torque is given, and the constructions above described show the variations in the speed.

Next, by differentiation, from the speed curve there is constructed a curve showing the accelerations of the speed. On another scale the acceleration curve shows the variation in the forces which are absorbed or given off by the rotating and oscillating masses at any change in the speed. If we add these acceleration forces to the effective torque applied to the shaft and subtract this sum from the theoretical torque we have in the remainder the forces used up in friction.

The difference measuring suggested assumes that all the values taken have been accurately determined.

Further discussion on this subject I will reserve for some later time.

DETERMINATION OF THE ENERGY ABSORBED BY HULL VIBRATIONS.

This is based on the fact that a considerable absorption of energy by hull vibrations occurs only at critical revolution speeds, where the energy absorbed is proportional to the square of the amplitude of the vibrations.

If the vibrations occur in the engine, the following method may be used.

All the shafting, including the thrust shaft, is uncoupled, and a test is made at no load with gradually increasing revolutions. As in Fig. 52, a curve is made for the horsepowers at no load and their corresponding revolutions. In this case the number of revolutions sometimes even exceeds the normal.

At critical speeds there enters a greater or less irregularity in the performance curve, somewhat like that in Fig. 52. An

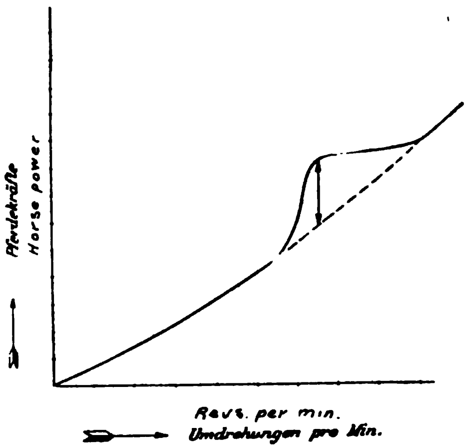


Fig. 52.

increase in the work done is here accompanied by scarcely any rise in revolutions, until the critical revolution speed is passed and exactly the opposite is true—a sudden increase in revolutions with a very small increase in work. If we prolong the curve we can pick off directly the horsepower used in generating the hull vibrations. If we measure at the same time the absolute vibrations we may obtain the damping constants for the vibrations.

This phenomenon was observed long ago in the case of bicycling. There are certain kinds of paving which, in connection with pneumatic tires, produce such vibrations that the speed can be increased only after large increase in power. The rider, indeed, finds this waste of energy highly inconvenient.

If the hull vibrations are caused by the propeller, not only will the I.H.P. be higher than the revolutions require, but the E.H.P. will be higher also. The irregularity of the curve is not here so noticeable because the power here absorbed by the vibrations is only a very small percentage of the whole E.H.P.

Considerable difficulty exists if the vibrations are due to both the engine and the propeller. It is quite possibly the

case that the engine and the propeller each cause strong vibrations, which often neutralize each other, giving the ship complete freedom from vibration. This depends, as Mr. Gumbel, Hamburg, first pointed out, on the size of the resultant forces, as may be observed from their phases.

A solution of the difficulty is possible from the fact that efficiency determinations go hand in hand with accurate pallographic measurements of the amplitude between the limits of vibration in each case. If the vibration is the same when the engine works with and without the propeller, the energy actually absorbed may be found as above described by reducing the horsepower.

If the vibrations are different after the propeller is coupled up, we are reduced to model experiments.

Further discussion of this subject would exceed the scope of the present article.

DESCRIPTION AND TEST OF 27-INCH CURTIS TURBINE FOR 50-FOOT U. S. NAVY CUTTER.

BY ENSIGN PAUL E. DAMPMAN, U. S. NAVY, MEMBER.

This turbine, intended for use in a 50-foot U. S. Navy Cutter, was built at the works of the Fore River Ship Building Company at Quincy, Mass. The contract requirements were that the installation of machinery intended for the cutter should develop 250 brake horsepower, as determined by test in the works of the company. After considerable delay, the turbine was completed on September 1, 1907, and received its official trial on September 11 and 12, 1907, at the works of the company.

DESCRIPTION OF TURBINE.

The turbine is of the usual Curtis type, with four ahead stages and one reverse stage. The pitch diameter of the wheels is 27 inches. The first ahead wheel and the reverse wheel each have four rows of rotating blades, while the remaining wheels have three rows of blades each.

Steam is admitted to the first ahead stage from the steam chest through seven nozzles. Of these, three are always open, the remaining four being provided with closing valves operated by means of a socket wrench. The astern steam chest has eight nozzles, none of which have closing valves. The stuffing glands on the shaft are packed with carbon rings. The weight of the turbine complete is 4,100 pounds. The designed horsepower is 250 when making 1,000 revolutions per minute, and using dry steam at 250 pounds gauge.

Two views of the turbine are shown. One shows the tur-

bine in the power house, complete for shipment; the other shows the details of rotor wheel and blading. The method of blading is interesting. A steel rod, of channel section, is punched with holes, spaced the distance apart of the blades. The blades, of extruded bronze, are driven into the punched holes and the shank riveted over. A series of saw cuts are then made in the under sides of the channel bar, enabling it to be bent to the curvature of the rotor wheel very much as a piece of molding is bent into a curve. The sides of the channel bar are then caulked into slots on the rotor, and the whole strengthened by means of a steel shroud ring, riveted over.

METHOD OF TESTING.

The turbine was mounted in the power house upon a cast-iron bedplate. Beneath this bedplate was a Wheeler surface condenser of 490 square feet of cooling surface. After condensation, the steam was delivered to a pair of measuring tanks, mounted on platform scales, by an Edwards single air pump of 10 inches stroke, 5-inch steam cylinder and 12-inch air cylinder.

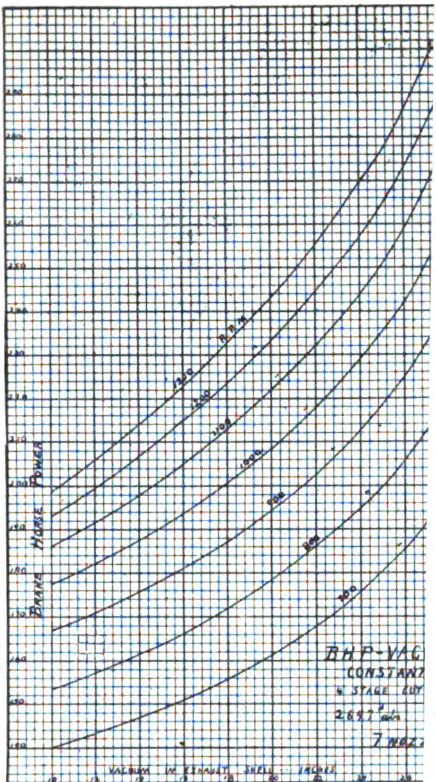
The turbine shaft was directly connected to a water brake, having an arm 5.252 feet in radius. Thus, brake pull, in pounds, times r.pm.. divided by 1,000 is equal to brake horsepower. The brake pull was measured by means of a platform scale.

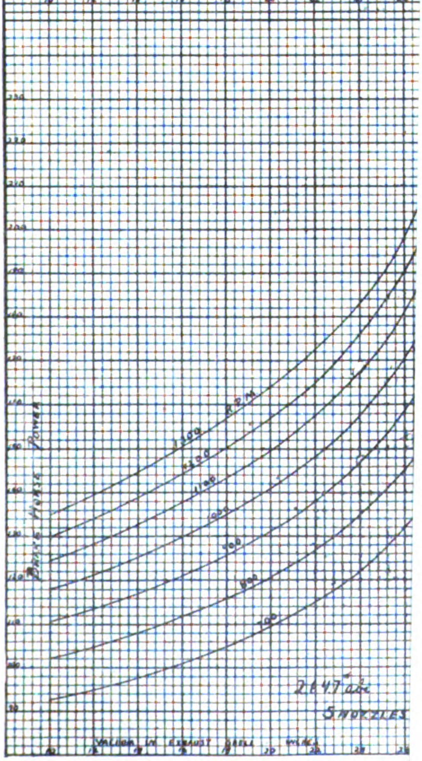
Steam was supplied for the test by a Babcock & Wilcox boiler, forming one of the Company's power plant, of the following dimensions:

Grate surface, 43 square feet; heating surface, 1,956 square feet.

This boiler was provided with a superheater, but only a part of the steam used was passed through the superheater. The object of having the steam slightly superheated was to be sure that the steam entering the turbine was dry and saturated.

In the separator, just before the steam reached the turbine,


BRAKE HORSE POWER
R.P.M.
1300
1200
1100
1000
900
800
700
VACUUM IN EXHAUST SHELL - INCHES
BHP-VAC
CONSTANT
7 NOZ


BRAKE HORSE POWER
R.P.M.
1300
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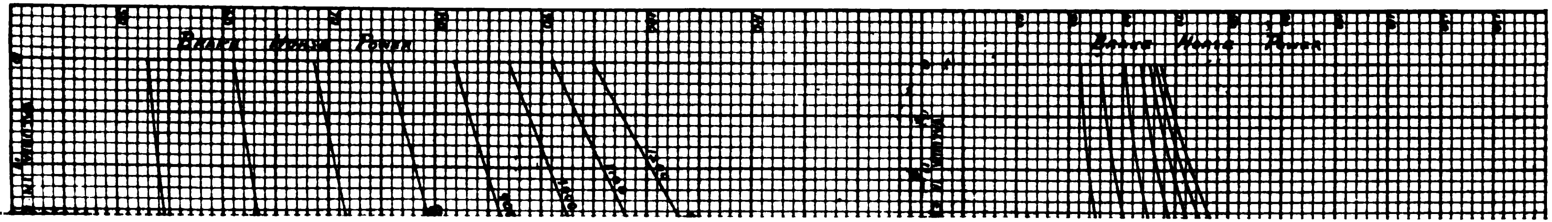
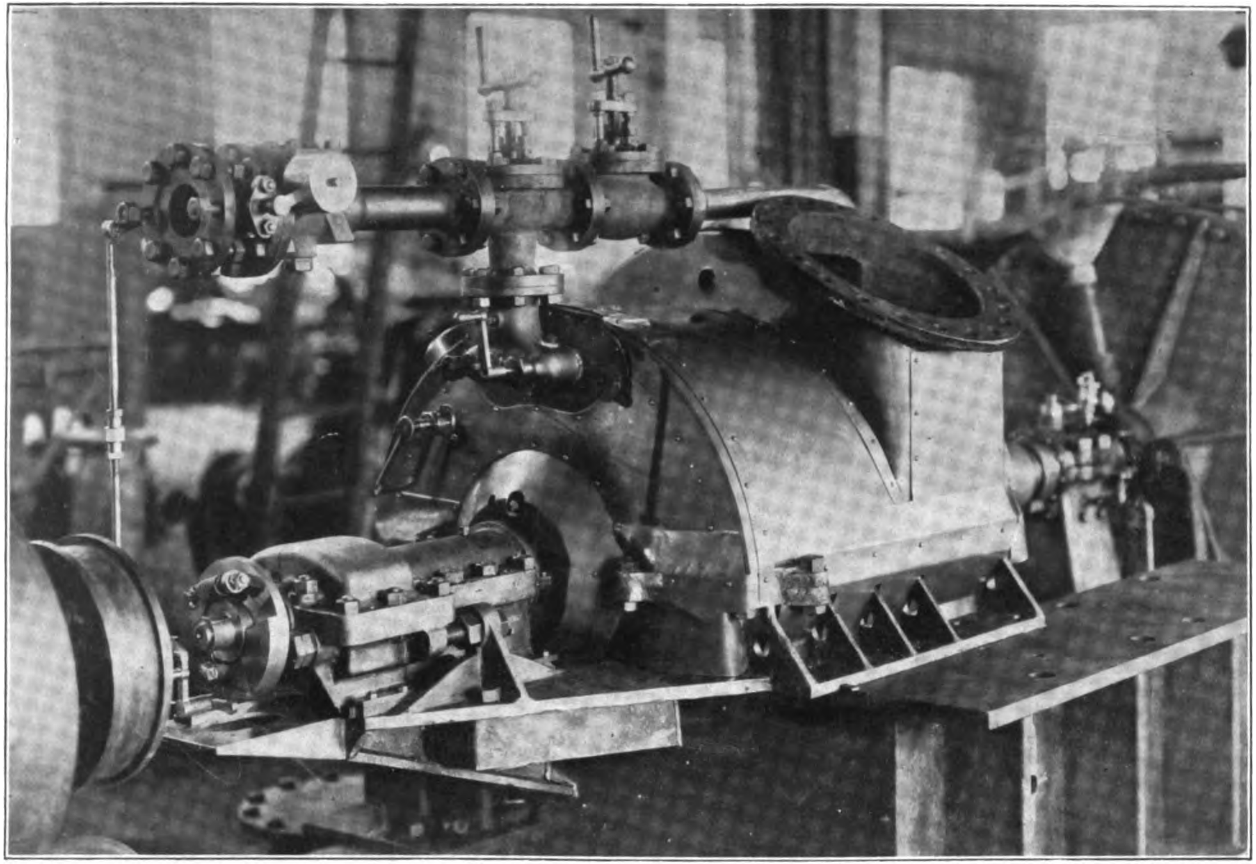
1000
900
800
700
5 NOZZLES

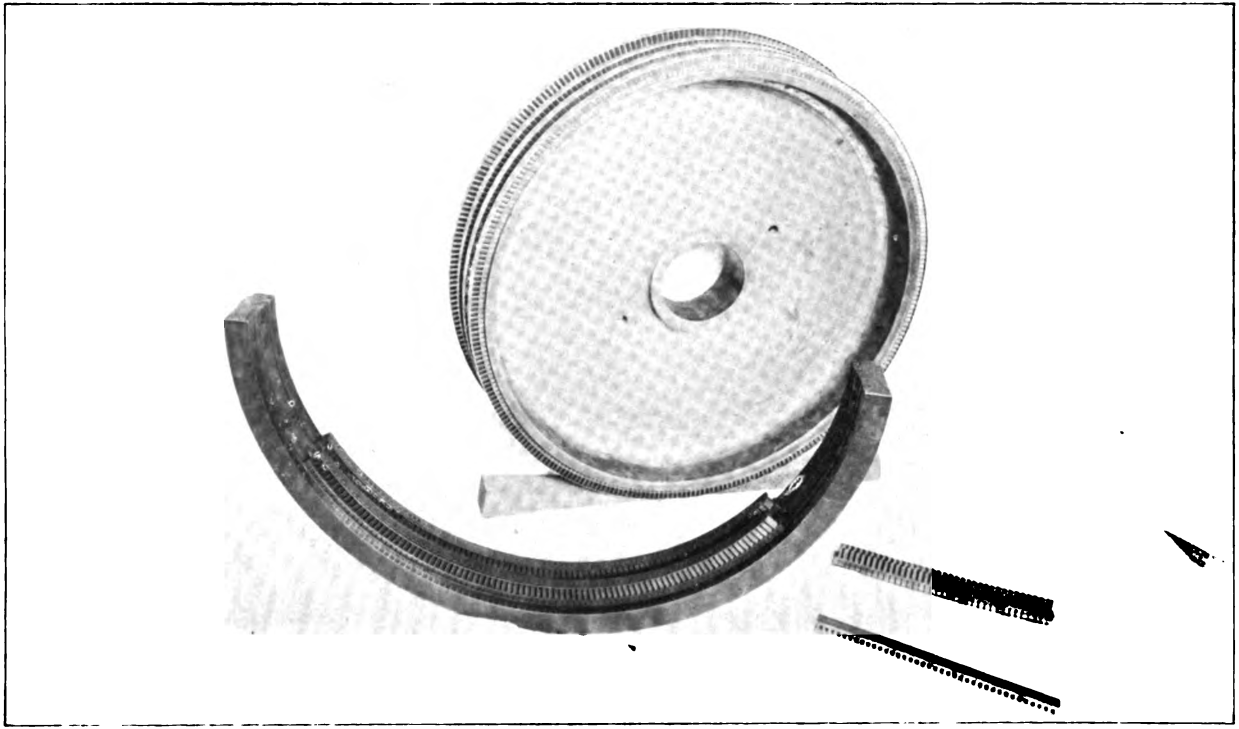
VACU



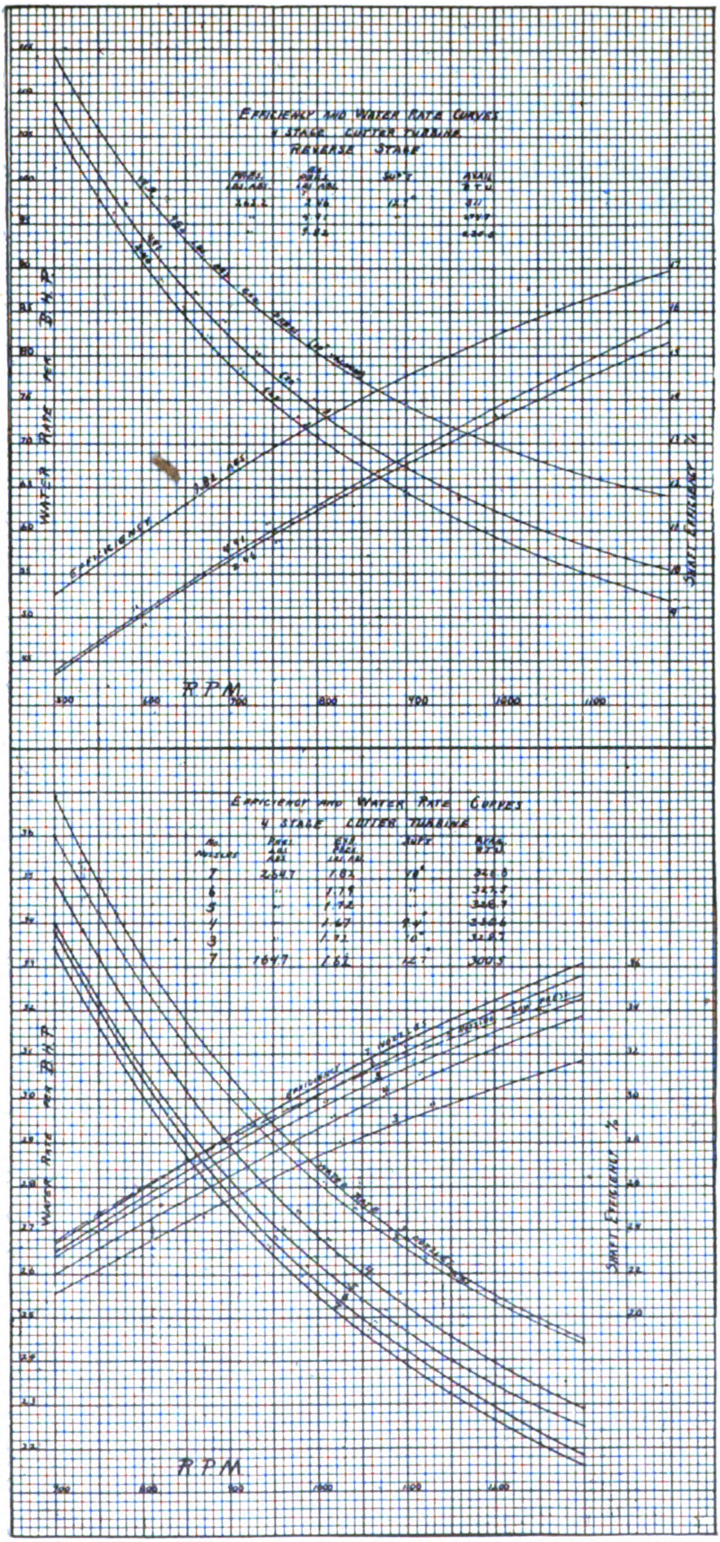
27 Inch Curtis Turbine, Complete.



27 Inch Curtis Turbine, Complete.



DETAILS, ROTOR WHEEL AND BLADING, 27-INCH CURTIS TURBINE.



Efficiency and Water Rate Curves, 27-Inch Curtis Turbine.

a cooling coil was inserted to give better control of the degree of superheat.

The pressure was maintained constant in the steam chest by means of a globe valve.

Pressures were measured by means of calibrated steam gauges. Vacuums were measured by means of mercury columns. Temperatures by means of high-grade glass thermometers.

The speed was controlled by regulating the amount of water fed to the brake. The speed was indicated by means of a tachometer, belted to the main shaft. The readings of the tachometer were checked at intervals of several minutes, by means of a small counter, thrown in and out by hand.

The highest vacuum which could be obtained was a little over 26½ inches. To regulate the vacuum, when a lower vacuum was used, air cocks on the condenser and exhaust chamber were opened.

Test runs were made with 3, 4, 5, 6 and 7 nozzles open, at three different speeds and four different degrees of vacuum.

A reversed test was made with three different degrees of vacuum. Also, using reduced steam pressure (150 pounds gauge), ahead tests were made, using the full seven nozzles and with three degrees of vacuum.

The object of making all these tests at different degrees of vacuum was so that some idea might be had of the performance of the turbine after it was installed in the launch, with a particular view to comparing the performance of the turbine-driven launch with engine-driven launches of the same type.

The boiler constructed for use in the launch is a water-tube boiler, of the Fore River type, of the following dimensions: Grate surface, 16.62 square feet; heating surface, 555 square feet; ratio of H.S. to G.S., 33.39 to 1.

The condenser is of the surface type, with a cooling surface of 203 square feet.

The air pump is a Blake Simplex Vertical Featherweight, with cylinders 4 and 8 inches in diameter, 6-inch stroke.

DATA SHEET—TRIAL OF 27-INCH CURTIS TURBINE.

Nozzles open.	Average steam flow per hour, pounds.	Exhaust vacuum, inches.	Chest pressure, pounds, gauge.	Chest temperature, Fahrenheit.	Pressure, first stage, pounds, gauge.	Pressure, second stage, pounds, gauge.	Vacuum, third stage inches.	R.P.M.	B.H.P.	R.P.M.	B.H.P.	R.P.M.	B.H.P.
Ahead 7	6,255	26.3	250	416	66.5	11.8	14.7	1,096	260.9	1,188	274.7	1,283	287.6
	6,255	24.0	250	416	66.5	11.7	14.5	1,091	244.2	1,185	256.9	1,282	269.2
	6,255	20.0	250	416	66.5	11.8	14.6	1,095	220.7	1,187	232 7	1,280	241.8
	6,255	15.0	250	416	65.0	11.3	11.15	1,092	200.6	1,191	210.1	1,284	216.2
Ahead 6	5,329	26.35	250	416	53.0	7.7	16.9	1,095	219.5	1,185	231.1	1,284	240.0
	5,329	24.0	250	416	53.0	7.3	16.9	1,093	200.8	1,189	213.2	1,280	218.4
	5,329	20.0	250	416	53.0	8.3	15.75	1,095	185.4	1,187	194.9	1,282	202.8
	5,329	15.0	250	416	53.0	8.2	12.0	1,095	169.4	1,188	176.4	1,281	182.2
Ahead 5	4,576	26.5	250	416	43.0	5.0	18.3	1,098	185.4	1,196	195.5	1,284	201.5
	4,576	24.0	250	416	42.5	4.8	18.4	1,094	166.7	1,192	175.7	1,283	180.0
	4,576	20.0	250	416	42.5	4.7	16.75	1,095	148.9	1,188	156.9	1,281	162.7
	4,576	15.0	250	416	42.7	4.8	12.6	1,093	134.4	1,186	141.8	1,279	145.6
	4,576	10.0	250	416	42.7	4.9	8.15	1,091	123.8	1,186	128.0	1,282	134.0
Ahead 4	3,748	26.6	250	415.4	33.0	11.8	20.2	991	138.5	1,149	152.7	1,281	161.9
	3,748	24.0	250	415.4	32.7	11.5	20.25	991	123.3	1,147	136.3	1,281	143.2
	3,748	20.0	250	415.4	32.7	11.5	17.7	993	110.9	1,150	121.3	1,281	129.0
	3,748	15.0	250	415.4	32.7	11.6	13.35	994	100.2	1,146	107.2	1,279	113.4
	3,748	10.0	250	415.4	32.7	11.6	8.75	993	90.3	1,149	96.9	1,279	103.5
Ahead 3	2,916	26.5	250	416	23.0	4.3	22.5	702	79.1	992	101.9	1,279	113.9
	2,916	24.0	250	416	22.35	4.2	21.65	702	70.1	992	89.1	1,280	103.4
	2,916	15.0	250	416	22.5	4.3	14.0	702	54.5	993	68.1	1,279	74.4
Ahead 7	3,915	26.7	150	378	34.0	7.0	20.0	993	138.7	1,150	151.1	1,279	159.4
	3,915	20.0	150	378	34.0	6.9	17.65	994	110.2	1,148	121.8	1,280	127.6
	3,915	10.0	150	378	34.0	7.0	8.6	992	88.9	1,148	95.1	1,280	100.0
Reverse. All nozzles (8)	6,072	25.0	247.5	418	...	...	...	799	85.4	995	102.1	1,191	116.4
	6,072	20.0	247.5	418	...	...	...	796	81.9	994	97.0	1,187	108.5
	6,072	10.0	247.5	418	...	...	...	797	75.4	993	86.3	1,188	94.7

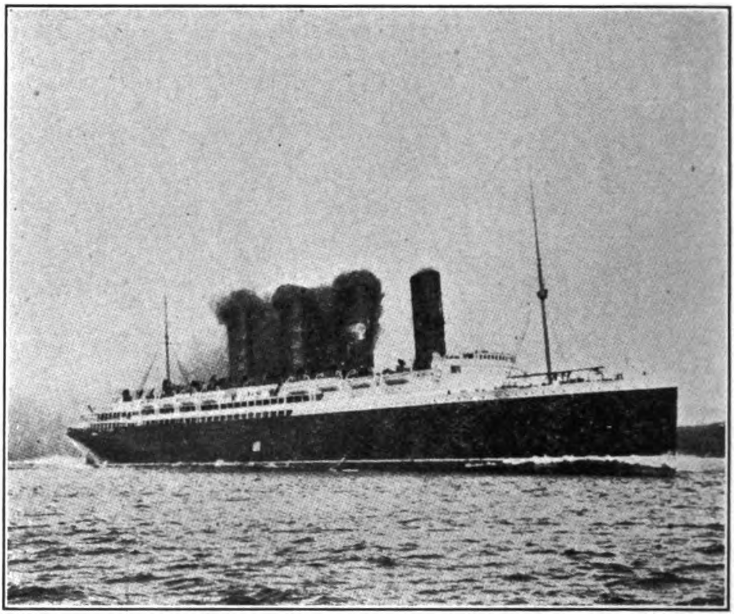
The feed pump is of the vertical, single-piston type, with cylinders 4 inches and 2½ inches diameter, and 4-inch stroke.

The circulating pump consists of a 10-inch impeller driven by a 2½-inch by 2-inch single engine.

Following are the weights of the machinery:

	Pounds.
Turbine,	4,100
Shafting,	587
Boiler,	5,262
Condenser,	1,138
Air pump,	280
Feed pump,	173
Circulating pump,	111
Total,	11,651

This does not include piping and connections, or water. Allowing 600 pounds for the necessary piping, and 2,000 pounds for water, the weight of machinery per designed B.H.P. comes out approximately 57 pounds.



THE CUNARD EXPRESS STEAMER "LUSITANIA."

THE CUNARD LINER *LUSITANIA.*

In her official trials, carried out under the direction of the technical staff of the Cunard Company and of the Admiralty representatives, the *Lusitania* met the most sanguine anticipations of all concerned. At a draught of 30 feet, she has steamed over 26 knots on the measured mile; on a 48 hours' sea run on long measured distances she has maintained a mean speed of 25.4 knots. The contract anticipated a speed of 24½ knots on the round voyage on the Atlantic.

The uprecedented length of the trial precluded "jockeying." The course of about 300 miles was traversed four times in alternate directions, so as to eliminate the influence of tide and weather. And thus any speed maintained on such a trial may be continued indefinitely, so long as coal and other supplies are available. It is unnecessary to say that the machinery worked satisfactorily. The general result stated carries conviction from this point of view. Before entering upon this, the most crucial test, the *Lusitania* had made several preliminary trials on the Clyde measured mile, not only to tune up the turbine machinery but to standardize the relation between revolutions, power and speed, so that a series of trials could be made to determine the coal consumption at various speeds.

For the economy tests the vessel was loaded to a draught of 32 feet 9 inches, equal to a displacement of 37,000 tons. Water and coal consumptions were taken while the vessel ran for six hours at speeds of 15, 18 and 21 knots respectively. The results were thoroughly satisfactory, but the data obtained were in connection with service requirements rather than scientific purposes. On the run to the Firth of Clyde, the starting point of the full sea-speed trials, the course measured out on the chart was between the Corsewall Light on the Wigtownshire

Coast to the Longship Lighthouse at Lands End, and this had to be traversed four times, alternately north and south. The compass bearings gave the distance, which aggregated about 1,200 miles. The weather was favorable, with cloudless days and starlight nights; but on both nights north-west winds freshened to forces of six and eight, and although this occurred when the vessel was steaming north, and somewhat increased resistance and slightly reduced speed, it brought consolation in the fact that it prevented fog. The feature of the trial was the uniformity of the speed on both runs south and on the two runs north, the latter being against the wind and tide. The course was divided into three approximately equal parts by the Fodling and Tuskar lights. Compass bearings taken at these intermediate points proved the uniform rate of steaming. The time taken on the runs south differed by only two minutes; further proof is unnecessary of the great regularity of steam supply or of turbine efficiency. The speed on four runs was: South, from Corsewall, 26.4 knots; north, from Longship, 24.3 knots; south, from Corsewall, 26.3 knots; north, from Longship, 24.6 knots; mean speed, 25.4 knots. We omit second-placed decimals, but, in any case, the percentage of error in observation is, with such distance, negligible.

This is a great performance: it exceeds by two nautical miles per hour any similarly long run made. The truest significance lies in the uninterrupted mechanical precision with which every unit of the machinery worked. The air pressure in the ash-pits of the boilers did not at any time reach the maximum of ¾ inch prescribed in the specification by the Cunard Company. The boiler pressure averaged 186 pounds per square inch, while the pressure at the receiver of the high-pressure turbines varied little from 150 pounds; at the low-pressure receiver it was 3½ pounds. The mean vacuum was 28.2 inches, with an average barometric reading of 29.8 inches. The mean revolutions of the four shafts were 188 per minute, and the power, according to the torsionmeter, was 64,600 horsepower. To those not versed in the details of steam-turbine performances the fact is illuminative. The circumferential or tip velocity of the ro-

tors of the low-pressure turbines was 150 feet per second, equal to over 9,000 feet per minute. The general procedure in the machinery department accorded with Atlantic practice, and the performance might to all intents and purposes have been two days' running, each equal to over 600 miles, on a voyage to New York. On returning to the Clyde the vessel proceeded on shorter distance tests between Corsewall and Chicken Rock, the southern extremity of the Isle of Man, and between the Holy Isle and Ailsa Craig, in the Firth, while following this were progressive runs on the measured mile at Skelmorlie, also on the Clyde; these ranged up to 26 knots, as already indicated in our introductory sentence.

The steering qualities of the vessel have also been tested. When steaming at 15 knots the rudder was put from amidships to hard over, both to port and starboard, in 15 seconds, and the full circle was completed in 8 minutes. Immediately before commencing to turn the engines were running at the rate of revolution which gave 15 knots. A careful record of revolutions was made on a time basis during the evolution, and it was found at the completion of the circle that the rate of revolution was then 70 per cent. of the rate at 15 knots. The final speed was thus assumed as 10.5 knots, the average speed 13 knots, and the diameter of the circle about 1,100 yards. This for a ship of this great length is a most satisfactory performance; the ship, at 22 knots, made the complete circle in 7½ minutes, with 15 degrees of helm. In ordinary steering the ship answered her helm very rapidly, according to the testimony of the pilot, and her swing was easily checked. Although the water was very fine, alike on the 36-hours' run around Ireland and on the 48-hours' trial on the deep-sea course, there was sufficient swell on the Atlantic in the first-named trip, and between the Tuskar and the Longship Lights on the subsequent runs, to cause pitching and rolling motions to be perceptible, and to afford opportunity for repeated records. In respect to these points a large number of observations gave the period of a single roll from side to side as almost exactly 10 seconds; a single pitch occupied 4 seconds. This

latter result calls for no remark, and the slow rolling indicates that, while the vessel has a satisfactory measure of stability, the long period of roll assures us that there will be the minimum of discomfort to passengers through this cause. The trials were finished August 1st.

As to vibration, it has for years been the custom at the Clydebank Works to make observations with Mr. Otto Schlick's pallograph, and the instrument has been used on every trial of the *Lusitania*. Mr. Schlick enunciated, in a paper read some years ago at the Institution of Naval Architects, a formula for determining approximately for various types of ships the first period of vibration and the slowest period of vibration of which the structure as a whole is capable. The correctness of this formula was borne out by the observations made on the *Lusitania*. The amplitude of vibration, although very small, was quite distinctly marked, and at a draught of 32 feet 6 inches and a displacement of nearly 37,000 tons, a first period vibration of a frequency of 62 per minute was clearly recognized on the pallograph records..

GOVERNMENT AND CUNARD LINE AGREEMENT; MAIL AND WAR-SERVICE SUBSIDY.

Under this agreement the Government provided a sum sufficient to pay for the new vessels, not exceeding £2,600,000, secured on debentures at 2¾ per cent. interest, while in addition £150,000 was to be paid per annum, on condition that the company would cause to be built, in the United Kingdom, two steamships of large size, capable of maintaining the minimum average ocean speed of 24½ knots in moderate weather. In the event of this speed not being maintained, and if the speed does not fall below 23½ knots a deduction is to be made from this annual payment by agreement. The clause in which this speed condition is set out is a matter of very considerable interest, and may here be quoted:

"If in the case of either of the two steamships mentioned in Clause 3 hereof, or any vessel substituted therefor, the company shall, before such steamship sails on her first voyage, fail

to adduce to the satisfaction of the Admiralty reasonable proof from trials that such vessels will be capable of maintaining a minimum average ocean speed of 24½ knots an hour in moderate weather, but shall prove to the like satisfaction that such vessel will be capable of maintaining an average ocean speed of not less than 23½ knots an hour under such conditions as aforesaid, then such deduction shall be made from the annual payment of £150,000 to be made by His Majesty's Government under the last preceding clause hereof as shall be agreed upon, or, failing such agreement, shall be determined by arbitration, by an arbitrator appointed by the Lord Chief Justice for the time being, and the decision of such arbitrator shall be final."

The minimum speed seems thus to be 23½ knots, although even then the ships may not be rejected; but the Cunard Company aim at, and will probably get, 25 knots, costly as that may be in respect of first charge, coal consumption, and upkeep. These two ships, in addition to carrying the mails and maintaining the prestige of Britain—which we regard as a very important commercial asset—are to be at the service of the Government in the event of war. To some it may seem remarkable that such an agreement should be necessary to secure the services of such vessels in emergency, and we may even have the naval critic urging that our cruisers ought to be equal in speed for any duty that the proposed Cunarders may fulfil. But many such forget the enormous difference between an Atlantic liner and a cruiser. In the first place such modern liners have a displacement twice that of the greatest cruiser built. In the merchant ship there is no armor to provide for, there are no guns to take, no ammunition nor naval stores, and the consequence is that the architect can allow for machinery something approaching double the weight per unit of full power. It follows that the same reliability in long-distance full-speed steaming cannot be guaranteed in cruisers. In other words, speed and weight of machinery are the main considerations in the merchant ship, whereas in the cruiser they are important, but probably equal only to gun power and armor protection. The Cunard ships cannot be equal to a cruiser in the latter

qualities for warfare; but they will be superior to similar ships which in time of stress can be withdrawn from the merchant service of other countries, and utilized for naval work.

It may be contended that patriotism alone ought to ensure that ships flying the British flag should ever be at the country's call; but the old order, which necessitated British ownership before the Union Jack could be flown, passed away in the adoption of the Limited Liability Act. Where the owning company is registered in this country under this Act the fact alone enables their ships to fly the British flag, even though not a penny is British capital, and not one shareholder—not even a director—is a British subject. This is not the place to enter into the question as to whether such a condition should be permissible. The British flag is certainly a great advantage to any ship in view of the maintenance of our naval power, and it is a question as to whether a change should not be made.

But the Cunard agreement with the Government involves the maintenance of the company as a purely British concern, one of the clauses of the agreement laying it down that the company will be under British control, "that no foreigner shall be qualified to hold office as a director of the company or to be employed as one of the principal officers of the company; and no shares of the company shall be held by, or in trust for, or be in any way under the control of any foreigner or foreign corporation, or any corporation under foreign control." The Cunard Company altered their Articles of Association to meet these conditions. While carrying on business to the best advantage, the company agree not to raise the freights or charges for the carriage of goods in any of their services; while no undue preference against British subjects is to be given. Plans of all ships to be built to steam 17 knots or upwards are to be submitted for approval; facilities are to be given for periodical inspection by the Admiralty, and for storing guns, ammunition, &c., at the ports; no chartering, except to the Indian Government, is to take place without notice being given to the Government, and the option to similarly charter. The Gov-

ernment is always to have the right of hiring the boats, the rates for such being: for vessels over 22 knots, 25 s. per gross register ton per month, and 5s. more if the company provide officers and crew; and for 20 to 22 knots, 20s., and 4s. more for staff; and for slower boats at less rates. In addition to holding the ships at the service of the Government, it has been prescribed in the agreement that all the officers, and three-fourths of the crew shall be British subjects, and that a large proportion shall belong to the Royal Naval Reserve. The ships are thus to be utilized as a great training school for British officers and seamen, and each month a record is to be made of the personnel with this point in view.

As to the mail service, the company shall, during the term of this agreement, convey, by means of mail ships from Liverpool (viâ Queenstown), or from Queenstown to New York, once in every week, on such day as may be provided, all such mails as shall for the purpose of such conveyance be tendered or delivered at Liverpool and Queenstown respectively, and shall employ as mail ships the fastest of the steamships for the time being belonging to or chartered by the company. For this mail service there shall be payable to the company a yearly sum after the rate of £68,000 per annum; but whenever in any one week more than 100 tons measurement (that is to say, 4,000 cubic feet) of parcel mails (exclusive of empty receptacles) in the aggregate are conveyed in either direction (whether by the mail ships or by any other steamships of the company), a further sum of 26s. 3d. is to be paid for every complete ton measurement; but the Postmaster General may, at his option, pay the rates of freight for the time being charged by the company on similar parcels to other companies or firms whose business is to carry parcels; but all parcels for which the said rates of freights are paid by the Postmaster General shall be carried by the company, subject to terms and conditions similar to those upon which the parcels of such other companies or firms are carried, and not under the terms and conditions of the agreement.

The loan of 2½ millions for the building of the two ships

is secured by a charge upon the whole of the company's assets, including all vessels built for or acquired by the company, so long as such steamships, or vessels, or any of them, shall remain the property of the company. The loan is to be repaid by the company by annual instalments, each of which shall be equal to one-twentieth of the total amount of the advance, the term of the agreement being for twenty years.

Under the trust deed under which the loan or debentures are to be issued three trustees hold office, the Government and Cunard Company's nominees electing the third. The company are further to issue to two nominees of His Majesty's Government one £20 share of the company, carrying the same voting power and other rights and privileges as an ordinary £20 share of the company; but, for the purpose of demanding a poll in respect of, and voting against, any special resolution involving any alteration of the company's articles of association, so far as respects the provisions referred to in the agreement, also carrying the following additional rights and privileges—namely, the right to demand a poll upon the occasion of any such special resolution, and the right to give against any such special resolution additional votes equal in number to one-fourth of the number of votes possessed by the company's share, stock or debenture holders for the time being.

THE CONTRACT; THE DIMENSIONS OF THE "LUSITANIA."

While the order for one of the two ships (*Lusitania* and *Mauretania*) was placed with Messrs. John Brown & Co., Limited, the other was awarded to Messrs. C. S. Swan, Hunter and Wigham Richardson, of Wallsend-on-Tyne, the machinery for the latter vessel being ordered from the Wallsend Slipway and Engineering Company, Limited.

The dimensions of the *Lusitania* are: Length over all, 785 feet; length between perpendiculars, 760 feet; breadth, molded, 88 feet; depth, molded, 60 feet 4½ inches; gross tonnage, 32,-500 tons; draught, 33 feet 6 inches; displacement, 38,000 tons; number of passengers—first cabin, 552; second cabin, 460; third cabin, 1,186; type of engine, Parsons turbine; number and type of boilers, twenty-five cylindrical; number of furnaces,

192; steam pressure, 195 pounds; total heating surface, 158,-350 square feet; total grate area, 4,048 square feet; draught, Howden's; total indicated horsepower (designed, 68,000 speed (designed), 25 knots.

One important feature dealt with in fixing the designs had reference to the use of the ships as cruisers or scouts in time of war, the machinery—which is almost entirely under the water line—has been so disposed in separate compartments, and with coal protection along each side, as to counteract, as far as possible, the effect of the enemy's fire at the water line. For purposes of attack the *Lusitania* will be provided with an armament as satisfactory as the armored cruisers of the County class, because on one of the topmost decks there will be carried, within the shelter of the heavy shell plating, four 6-inch quick-firing guns attaining a muzzle energy of over 5,000 foot-tons, while on the promenade deck on each side there will be four more guns on central pivot mountings, also able to penetrate 4¾-inch armor at 5,000 yards range, and 6-inch armor at 3,000 yards range. With the great speed, which can be maintained for three or four times the period that any modern cruiser can steam even at only 21 knots, and with the careful subdivision for protection and their satisfactory offensive power, the *Lusitania* and her consort may be regarded as most effective additions to any fighting squadron. Their advent is therefore a great advantage from the point of view of British sea power.

The rudder and steering gear are all placed well below the water line. This is a most important point in respect of protection, should these vessels be ever impressed into the national service. The stern has been suitably shaped in the *Lusitania* to enable this object to be accomplished satisfactorily.

THE ADOPTION OF STEAM TURBINES.

The second problem in design was the question of the type of propelling machinery to be adopted. The power of the machinery, for the dimensions and form evolved in the Government tank at Haslar, was 68,000 indicated horsepower. Ex-

perience has produced an exact rule as to the efficiency of tank boilers as steam generators, and thus the capacity of the boiler installation was a more or less fixed quantity; but it was important to determine whether the steam efficiency of the steam turbine or of the reciprocating engine was greater. Even in a ship where speed is the first desideratum it is incumbent upon the designer to aim at economy, alike in first cost, working expenses, and maintenance charges. Three years ago, when the decision had to be arrived at, there were comparatively few data even as to the steam consumption of turbines, and less as to their durability; but the Cunard Company, with that wisdom which has brought them to the front rank, decided to appoint a commission of experts to investigate and report upon the whole question. The company were equally felicitous in their selection of the experts.

This Commission entered upon their work with great care, conscious of the responsibility resting upon them: on their decision the success of the new ships largely depended. Admiral Oram, who, as Deputy Engineer-in-Chief of the Navy, has tackled the question of turbine economy, alike from the scientific and practical standpoint, with enormous advantage to the Navy, was able to put before the Commission very important results as to the performance of destroyers. The late Mr. Brock and Mr. Parsons assisted the Committee with the tests of Channel steamers into which their respective firms had fitted turbine engines. The resultant data, and a careful consideration of the performance of turbines ashore, encouraged the Committee to make the bold step of recommending for these huge liners the new prime mover. Events have since justified this intuition, and the consequence is that the Cunard liners are in the forefront, not only in speed, but in the method of attaining it. The constructors of the machinery, in accepting the contract for turbine machinery, with the heavy guarantees attached, also displayed characteristic enterprise.

The Clydebank firm entered upon their work in a thoroughly practical way, laying down immediately a complete turbine installation, where, by dynamo-metric means, they were able

to test the power, while at the same time measuring the water consumption. Many questions affecting the details of design were similarly experimentally determined, and the result was that they induced the Cunard Company to at once adopt turbine machinery in a large steamer of 20 knots speed, the order for which had just been placed with them. We have already dealt very fully with the machinery in this ship, the *Carmania*, and, at the same time, have described the experiments undertaken and the modifications made in the turbine, so that it is not necessary here to say more regarding this phase of the subject. It may, however, be indicated, as suggestive of the completeness of the experimental plant, that it was subsequently fitted to a Clyde steamer direct from the testing house, and has since given very satisfactory results, corroborative of the high efficiency attained at the works.

FORM OF STERN AND NUMBER OF PROPELLERS.

The adoption of turbines was immediately followed by very careful experiments as to the form of stern suitable for four propellers, and as to the proportions of propellers. Four screws were imperative, whether turbine or reciprocating engines had been adopted, because great ingenuity had to be devised in connection with the distribution of power in the cylinders of the piston engines immediately preceding Atlantic liners. In the *Campania* and *Lucania*, where the total power was 30,000 indicated horsepower, two sets of engines were found sufficient, each having three cranks with five cylinders, the high and low pressure being arranged tandemwise, with the intermediate cylinder in the center. In the next large ship, the *Kaiser Wilhelm der Grosse*, there were two sets of engines of the four-cylinder triple-expansion type, each cylinder working on a separate crank; although the power was the same, with a slight increase of steam pressure, the low-pressure cylinders were 96.4 inches in diameter. The Vulcan Company, in their succeeding ships, adopted various systems, and in the *Kaiser Wilhelm II* fitted four sets of engines, two on each shaft. Each set of engines had three cranks, and the tandem system

was again adopted for high and first-intermediate cylinders, but the diameter of the low-pressure cylinder was increased to 112.2 inches. This latter dimension has been exceeded only in one or two instances, in compound-paddle engines. The six cranks probably improved the uniformity of turning moment. With tandem cylinders the difficulties of balancing the moving parts were necessarily increased, and it is probable that, as a consequence, some portion of unnecessary weight had to be carried. In any case, the transmission of 20,000 indicated horsepower through a single shaft, setting up enormous torque, imposed a very severe condition even upon the best of steel makers, and those responsible for the design of the machinery in these new liners were, therefore, well advised in aiming at a reduction to about 16,000 or 17,000 horsepower through any one shaft.

The adoption of four units of machinery and four screw propellers enabled the machinery to be made in two completely separate sets, one to starboard and the other to port, just as in a vessel with twin-screw reciprocating machinery; a distinct advance upon the triple-screw arrangement in former turbine ships, where such complete independence is unattainable. The division of the power into two complete and independent systems follows the course pursued for so many years in the Royal Navy, whose lead is now so generally adopted in this respect in the mercantile marine. It has in this case enabled the engine rooms to be well subdivided by watertight bulkheads, and the advantages in general secured have been obtained without the sacrifice of a single point of any importance.

As in earlier Clydebank ships, the rudder, of the barn-door type, is supported for about two-thirds of its depth. Immediately forward of it, on each side, are the two inner propellers, the shafting for these being entirely borne within the ship, the framing of which was bossed out, and strongly supported by heavy webbing, as explained later. The forward propellers are about 70 feet ahead of the inside screws, and here also the frames are carried by heavy webs. Owing to the great beam of the ship and the very fine run, the blades of the outside propellers do not project beyond the beam line, while, at the same

time, all the propellers work in free water, and provision has been made for a satisfactory clearance between the propellers and the skin of the ship.

Another question affecting the efficiency of the propelling machinery had reference to the revolutions at which the propellers were to be run at full speed. In Channel-steamer work, and in small craft generally, a high rotating speed has been adopted, largely as a matter of necessity. Where weight is limited, it is important to minimize the dimensions of the turbine, and the velocity of steam being constant, the revolutions are of necessity high. The contention has been made—not always with due regard to experience—that the small high-speed propeller involves some loss in efficiency, especially in a seaway, and, further, detracts from the astern speed, and, indeed, from the general maneuvering power of the ship. Be that as it may, it was decided in the Cunard liners, after very careful consideration, to attain the full speed, with the propellers making about 140 revolutions per minute, and the turbines were proportioned to suit this speed. The peripheral speed being practically more or less constant, owing to the velocity of steam, and only affected by the angle or curvature of the blades, it became necessary to adopt turbines of very large diameter. Thus the rotor drum of the high-pressure turbines is 96 inches in diameter, and that of the low-pressure turbine 140 inches, the blades ranging from 2¼ inches to 22 inches in length. The result is to permit the use of a propeller of a diameter and pitch which will certainly remove any question as to relative efficiency, under normal conditions, as to maneuvering power and astern speed, and also as to the influence of head seas. From these points of view, the performance of the ship and the machinery will be watched with very careful interest; although the results already attained with the *Carmania* prove that the line of reasoning which has actuated the designers of the propelling machinery of the *Lusitania* is correct, and therefore there is every likelihood of a full practical success.

THE VENTILATION OF THE SHIP.

The thermo-tank system, as recently introduced by the Thermo-Tank Ventilating Company, of Glasgow, has been adopted in connection with the ventilation of the ship. This system aims especially at ensuring to all the living quarters of the ship a continuous supply of fresh air, which is not only warmed to the requisite degree, but is also humidified, so that none of the bad effects of over-drying can be felt. In cold weather the warmed air is discharged, through a regulated louvre, into each apartment, near the level of the ceiling; as it cools it gradually sinks to a lower level, carrying with it any carbonic-acid gas to the passage ways, where means are provided for allowing it to pass outside. In warm weather, or when heating is not necessary, the reverse action takes place, as the louvres near the ceiling constitute the exhaust, with the result that the warm impure gases leave the top of the room, and fresh atmospheric air comes in at the floor level.

The thermo-tank generally consists of an electric motor operating a fan which discharges air to the outside of a tube heater. The air then passes through the tubes, and comes in close contact with the heater surface, flowing thence to the main distributing trunks. Two valves are used for controlling the passage of air; one for regulating the temperature, while the mushroom valve on the top is provided for the exhaust air. It will be noted that the air passes round the outside of the heater on its way to the tubes, so that the loss from radiation is very small, the outer casing of the thermo-tank being quite cool on all occasions. The heater is warmed by steam from the main boilers, entering at the top, with an exhaust at the bottom. The pressure of steam is reduced to about 30 pounds, and a relief valve is fitted to blow off at from 80 pounds to 100 pounds pressure. The heater and all its connections are tested to the full boiler pressure. The air is humidified by means of a special valve admitting steam in a fine spray, by means of small needleholes in a copper hoop surrounding the heater. Tests carried out to compare the efficiency of the thermo-tank system with that of the ordinary heaters show that where the

steam-heated system took three hours to attain a given temperature, the thermo-tank only required fifteen minutes. The consumption of steam is small, as all the heat is abstracted, only water being drained off to the feed tank.

The first-class accommodation is connected to twenty-four thermo-tanks, which are arranged principally on the boat-deck houses. The second-class accommodation is connected to nine thermo-tanks, the third-class to eleven, and the officers' and crew's accommodation to five, these being arranged mostly on the top of deck houses, &c. The thermo-tanks in the fore end of the ship are placed between decks, and obtain their supply of fresh air from the after end of the navigating bridge, so that in this way a continuous supply of fresh air is ensured in the worst weather, there being no cowl heads or openings forward of the flying bridge. Although the thermo-tanks are arranged principally on the top of the boat-deck houses, the fresh-air supply is obtained from gratings opening out on the promenade-deck shelter. This has been done so as to avoid the smell from galleys, w.c.'s, &c., which all exhaust above the boat-deck houses. When the thermo-tanks are exhausting, of course, the cowl head provided for the purpose can then be used.

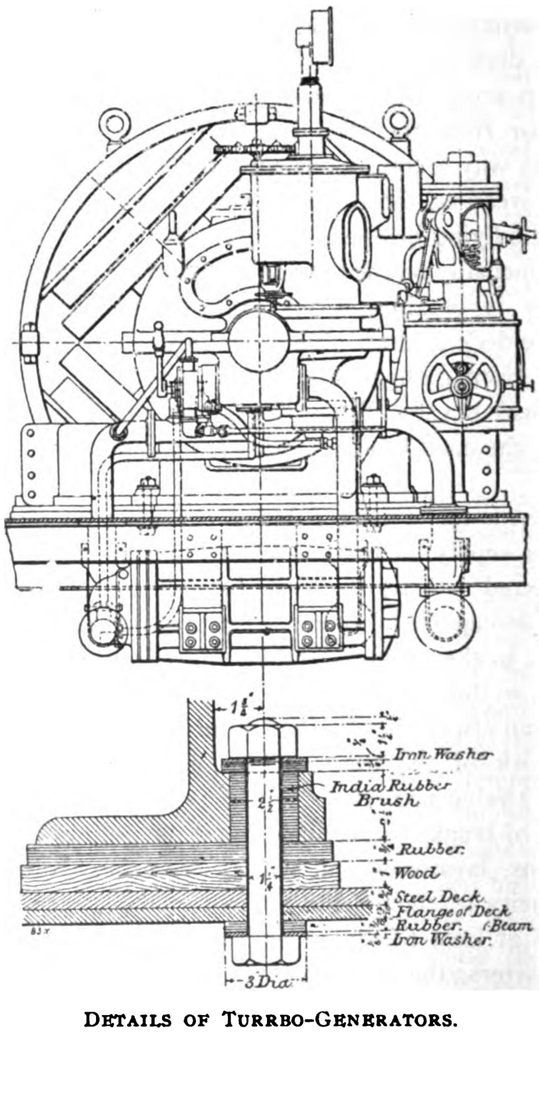
The thermo-tanks are capable of changing the air, either by exhaust or supply, in the various compartments to which they are connected at least from six to eight times per hour, and they are also capable of maintaining a temperature of at least 65 degrees F. in the coldest weather. In addition they are interconnected, so that in case of the breakdown of any thermo-tank, a supply can always be obtained from another.

The work of the thermo-tanks is further augmented by means of twelve powerful exhaust fans, which are connected by means of trunks respectively to all the galleys and pantries, bath rooms, lavatories and w.c.'s, these fans being of sufficient capacity to change the air in the above-mentioned compartments at least fifteen times per hour. In addition to the living quarters, the holds and other compartments, forward

and aft, are also mechanically ventilated, so that all natural ventilation requiring cowl heads, or openings through decks, has been dispensed with.

ELECTRIC LIGHTING.

An extensive electric generating station is arranged on a flat deck abaft the engine rooms. There are four generating



DETAILS OF TURRBO-GENERATORS.

sets, all alike, and each of 375 kilowatts capacity, the voltage being 110 to 120. The prime movers are Parsons turbines, and it is interesting to recall that the Clydebank firm was among the first, if not the first, to fit turbo-generators on ships, and on that occasion the vessel also was for the Atlantic trade. The turbines were designed to give full load when exhausting into a back pressure of 10 pounds. They run the dynamos at 1,200 revolutions per minute; but an overload of 10 per cent. for two hours is provided for. Each dynamo is shunt wound. The armature is of the surface drum-wound type, with one turn per section. No relative movement between the conductors and commutator bars is possible, but as an addi-

SUMMARY OF OFFICIAL TRIALS OF TURBO-GENERATORS.

Number of machine	1034.			1077.	1078.			1079.
Date of test	4-8-06	4-8-06	4-8-06	14-8-06	13-8-06	13-8-06	13-8-06	14-8-06
Load	Full	Three-quarter	Half	Full	Full	Three-quarter	Half	Full
Stop-valve pressure	167	167	173	166	158	161	160	164
Barometer, in inches of mercury	29.77	29 77	29 77	29 54	29.4	29.4	29.4	29.54
Back pressure, in pounds.	5.0	5.0	5.0	4.95	4.86	5.0	5.0	5.0
Speed, revolutions per minute	1,200	1,200	1,200	1,200	1,200	1,200	1,200	1,200
Voltage	111	111 2	113.6	115.2	107.0	109.8	112.3	114.5
Average kilowatts	373.27	288 9	188.42	375.38	371 75	285.28	188.3	373 31
Field volts	87 6	88.73	86 06	92 46	87.5	89 6	90.6	92 08
ampères	33.31	31 15	30 9	30.76	30.8	30.8	30.8	31.5
Average quantity of water per hour, in pounds,	17,831	15,017	11,419	17,301	17,888	15,104	11,649	17,546
Water consumption per kilowatt-hour, in pounds,	47 76	51 97	60.60	46.08	48.14	52.94	61.86	47.0

tional safeguard the connections between these are made flexible. Special driving horns in the ends of the core are provided. The insulation of the whole machine was tested with an alternating pressure of 2,000 volts between the conductors and the frame, and the insulation resistance after the above test was to be not less than one-half of a megohm for the whole machine, one megohm for the armature winding, one megohm for the field winding, and one megohm for each of the brush holders.

Tests were made of the various turbo-generators at the Heaton-on-Tyne Works of the constructors—Messrs. C. A. Parsons & Co., Limited—and the results for all four engines are given in the table. It will be noted that at half load the water consumption was in one case 60.60 pounds, and in

another 61.86 pounds per kilowatt-hour; at three-quarter load the consumption was 52 pounds to 53 pounds, and at full load from 46 pounds to 48 pounds, the back pressure in each case being about 5 pounds.

REFRIGERATING MACHINERY.

Two complete and independent installations of refrigerating machinery are fitted on board the ship, one for the preservation of the ship's provisions, and the other for the carriage of perishable cargo. Both have been constructed by the Liverpool Refrigeration Company, Limited, and must be described together. The ship's provision machine is situate near the forward end of the turbine-engine room on the main-deck level, and the chambers on the lower deck, port and starboard sides, some distance forward of the machine. These chambers have been insulated at the Clydebank Works with granulated cork in combination with specially-treated damp and rot-proof paper, with linings of white-pine boards. The chambers are divided into compartments for beef, mutton, poultry and game, bacon, milk, fruits and vegetables, and ice, and the wine and beer and spirit chambers are also lightly insulated and cooled to a suitable temperature.

The chambers have a total capacity inside insulation of about 13,000 cubic feet, and, in addition, there is a large cold larder on the upper deck, besides cold boxes in the first and second-class bars and still room. There is also an installation for the supply of cooled water for drinking and other purposes. In this connection every possible requirement has been thought out and arranged, not only for the preservation of the perishable provisions in bulk, but also for the convenience of the catering and culinary departments generally.

The installation is of the carbonic-anhydride type and consists of a horizontal compound duplex machine, mounted on a cast-iron box bed, which is divided by a longitudinal bulkhead into two portions, each of which contains an independent set of gas-condenser coils. These coils are of special soft-iron lap-welded tube, galvanized on the outside. The compound

steam cylinders drive from their tail-rods two horizontal double-acting CO_2 compressors. The crank shaft runs in four bearings, and is in two portions, coupled in the center and with a distance plate between the faces of the coupling. A neat arrangement of steam valves is fitted, so that the engine can work compound or independently as two high-pressure engines. By taking out the coupling bolts and distance plate each side of the machine can be run quite independently of the other. Cross-connections are provided, so that either compressor can deliver into either or both gas condensers, and the machine is the full equivalent of two independent machines combined on one base.

The evaporator is of the vertical type, the shell enclosing two independent nests of circular coils, one coupled to each compressor, with cross-connections, the same as for the condensers. Two horizontal duplex brass-fitted brine pumps circulate the brine, and a third, smaller and independent pump is provided for the special duty of pumping the brine supply to the cold larder and refrigerators in the saloon bars and other places independently of the main pumps. The whole brine distribution forms an entirely closed system. The brine is drawn from the evaporator and delivered into a distributing header, with valves and connections leading to the various pipe sections in the cold rooms. After passing through these the brine returns to a similar collecting header, and thence back to the evaporator, there being no open brine tank whatever. All the chambers are cooled with galvanized brine piping, arranged to suit various temperatures required in the several compartments, each of which is regulated independently of any other. The installation is, we believe, the largest and most complete of its kind.

The cargo-refrigeration plant is situate on the shelter deck, starboard side forward, just abaft the forward funnel hatch. There is an extensive range of cold chambers on the orlop decks forward. These have been insulated at the Clydebank Works with granulated cork in a similar manner to the provision rooms, already described. There are six chambers in

all, the largest ones being divided by longitudinal central bulkheads. They have all been fitted for the carriage of frozen meats and poultry, cheese, bacon, butter and fruits, and particularly for the carriage of chilled beef. The compartments are quite independent of each other, and can be supplied with brine for cooling at any temperature suitable to the cargo carried. The brine circulates through galvanized wrought-iron piping, and the chambers are fitted with meat rails and removable hooks for hanging chilled meat, and also removable side tables for the stowage of forequarters. As the machinery is at a considerable distance from the chambers, a brine distribution house has been fitted on the shelter deck near to the chambers, from which the regulation and distribution is controlled.

The machinery in this case also is of the carbonic-anhvdride type, and special care has been taken to ensure silent running. The plant is in duplicate throughout, and is electrically driven. There are two horizontal gas compressors, each direct coupled to a powerful electro-motor. These motors have been specially designed and constructed for the purpose by Messrs. Boothroyd, Hyslop & Co., of Bootle, and are so arranged, by means of shunt regulation, that they can run at any desired speed from 40 to 110 revolutions per minute. The speed can be regulated with absolute ease by the turning of one hand-wheel only, the motor running at the same speed as the compressor, and no gear wheels whatever are used. The compressors—Webb and others' patents—embody several new features, which it will be of interest to mention. The outer casing, of soft cast steel, encloses and supports a liner of hard close-grained cast iron, which forms the working bore of the cylinder, but is easily withdrawable from either end of the casing. Two forged-steel headers, carrying the valves, are bolted, one to each end of the casing. The one at the front end is fitted with the stuffing box and gland, and that at the back end with the plug cover. The piston is fitted with metallic packing rings of special metal, very accurately turned and finished, and held in place by a patent split junk-ring head, which, while doing away with all

screws, keys and pins, absolutely secures the rings in place, so that they cannot get adrift as long as the piston is within the cylinder bore. The gland is also fitted with a particular form of metallic packing, and no leather cups are used. The valves and seats are of special hard steel; the valves lift vertically, are of large area, and have no springs whatever. The compressors are constructed so as to be capable of long continuous runs without stop; the absence of leather cups entirely does away with the necessity for frequent renewals of the packing.

The gas condensers are independent, are of the vertical type, and consist of soft lap-welded coils of wrought iron, galvanized on the outside, and contained in galvanized wrought-steel shells. The evaporators are similarly constructed to the condensers, ample facilities being provided for easy access to the coils for cleaning.

Two high-lift Gwynne centrifugal pumps circulate the brine. These are direct coupled to variable-speed electro-motors by the same makers as the main motors, so arranged that the speed of each pump can be regulated to suit the resistance to be overcome, this resistance varying somewhat, according to the number of chambers in use, and the quantity of brine being circulated. We have already mentioned a large variety of goods that may have to be carried in the chambers, some frozen and others chilled. Frozen goods, of course, require brine at a low temperature for circulation through the cooling pipes in the chambers, but when carrying chilled beef, for example, brine at a temperature suitable for frozen goods is altogether too cold, and, if circulated, would rapidly freeze the quarters, especially those stowed nearest to the pipes. The temperature must be regulated with great accuracy. The same remark is applicable to certain fruits and other chilled produce. A brine supply at an accurate and easilv regulated temperature is, therefore, of great importance, and in the installation under review special means have been provided so that this can be secured. This warmer brine is circulated by an independent pump, and in the distribution room mentioned above special

duplex headers have been arranged so that either the coldest brine or the warmer brine can be supplied to the cooling pipes in any one or more chambers, according to the cargo carried, whether chilled or frozen.

Webb and others' patent brine attemperator consists of a simple three-ported slide valve, enclosed in a cast-iron casing, and attached to a screwed spindle with a hand-wheel, so that the movement of the valve over the ports can be accurately regulated as required. The valve is kept up to its face by a suitable coach spring. A branch from the cold-brine supply main is coupled to one end of the valve casing, and the return warmer-brine main is coupled to the other end, the mixed or attemperated brine escaping through the central port and pipe, which is connected to the warmer-brine pump. An overflow pipe is connected back to the evaporators.

In working the apparatus, the warmer-brine pump draws from the mixing or attemperating valve chamber, delivering to the headers already mentioned, thence through the pipes in the chilled chambers back to the attemperator, there to be mixed with any given proportion of the coldest brine necessary to lower its temperature to the required degree. A suitable and specially-constructed pyrometer indicates the temperature, both the valve handle and thermometer being carried outside the insulated brine room. The temperature can be regulated by means of the handle controlling the valve exactly as required, higher or lower to suit, and the control is positive. The brine circulation generally, as in the provision plant, is an entirely closed circuit. There are no brine tanks, except a small one for mixing brine in the first instance, for charging the machine or for adding a little from time to time. Any little air or foul gas in the system is automatically disposed of through a small vent pipe carried outside from each evaporator. Though there are a large number of independent circuits, no difficulty whatever is experienced in regulating each exactly as required. With the closed circuit there is no difficulty with air locks or aeration of the brine, and the system is surprisingly simple, clean and easy to work.

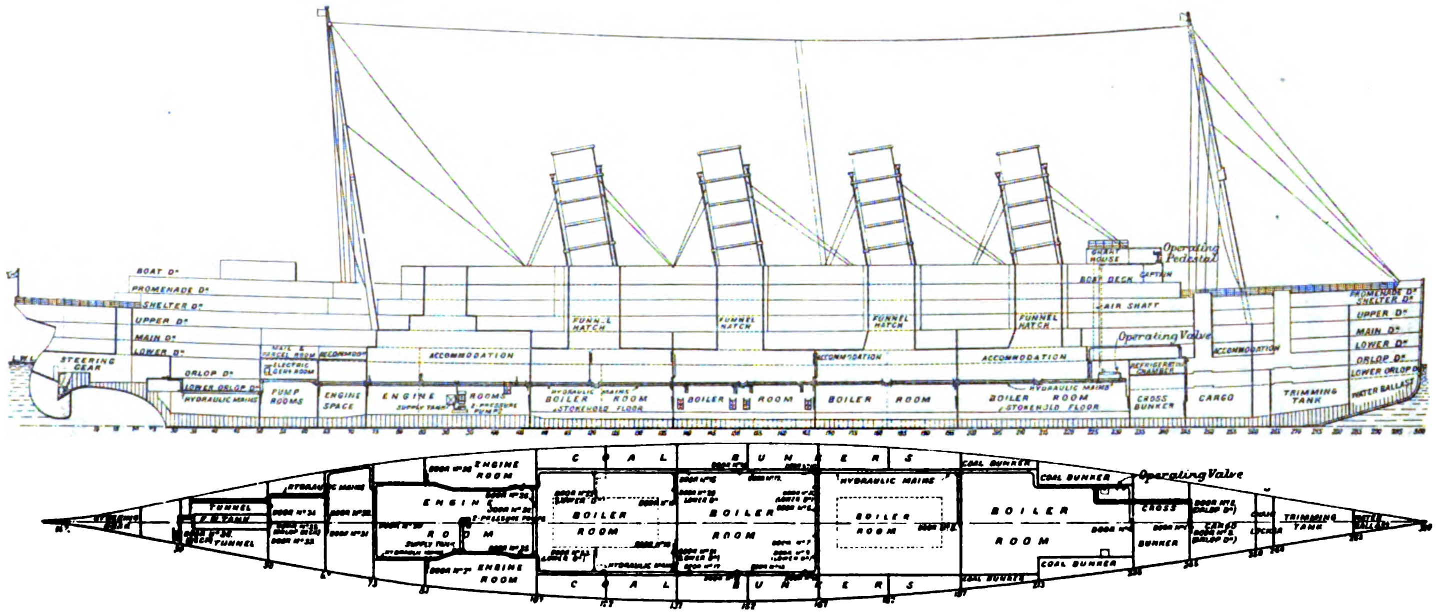
In both the cargo and provision installations the cold parts of the plants are not individually lagged, but are placed together in a well-insulated chamber or brine room, where they are always accessible, without the necessity of removing any lagging. The whole of the machinery, piping and fitting out of both plants has been carried out by the Liverpool Refrigeration Company, Limited, Liverpool, the installation being the latest of a large number of refrigerating plants fitted by this company for the Cunard Line.

SAFETY OF THE SHIP.

In describing the construction of the ship we have referred to the arrangement of bulkheads to ensure the safety of the vessel should any collision or other mishap cause the entry of the seawater. In this respect she will be much safer than almost any other vessel. It will be obvious, however, that if intercommunication is to be maintained freely between the various machinery and other compartments, these bulkheads must be pierced; otherwise serious and innumerable difficulties present themselves. Experience has proved that the absence of such doors is impracticable, and it is necessary to have effective, reliable, and above all, a quick method of closing the doors. This has been accomplished in the *Lusitania* by means of the well-known Stone-Lloyd system, constructed by Messrs. J. Stone & Co., Limited, of Deptford. After careful consideration they have preferred hydraulic power for actuating the doors, discarding steam, electricity, and compressed air.

Hydraulic pressure is supplied to each of the doors by a pressure main which runs round the vessel. Pressure is maintained by two steam-driven duplex pumps placed in the engine room, and continually under steam. A branch from the pressure main feeds an operating valve, which is placed on the casing of the forward boiler room, so that the pressure may be led into a small pilot main, called the "closing main," which also runs round the vessel to serve the doors.

The operating valve is connected by telegraph wires and chains to a pedestal on the bridge, so that pressure may be



ARRANGEMENT OF BULKHEADS AND WATERTIGHT DOORS,

led into the closing main, either from the operating valve itself, or by the pedestal on the bridge.

The doors are of the ordinary wedge type, and are formed of steel plate, suitably stiffened. They are each operated by an hydraulic cylinder, having two pistons connected by a rack, which gears with a pinion carried by a cross shaft; the shaft prolonged forms the door shaft, which, in turn, carries the driving pinion, gearing with the rack on the door. When space prohibits the fitting of the cylinder in the immediate neighborhood, intermediate shafting, with bevel wheels, can be arranged, and the cylinder can be placed in any convenient position.

The pistons are of slightly different sizes, so that a larger force is available to open the door, or bring it off its wedges. But the successful working of the system may be said to lie with the controlling valve, which is placed at each door. It consists of a tubular ram, which slides in a casing and is operated externally by a lever. The ram carries at its center an ordinary slide valve, which slides over three ports. These ports lead respectively to the opening and closing ends of the cylinder, and to an exhaust main, which runs round the vessel, and delivers into the supply tank. The pressure from the main is constantly behind the slide valve, so that, according to the position of the latter, the pressure flows either to the closing or to the opening end of the cylinder, the other end meanwhile exhausting. The ram is moved by a controlling handle, which is connected by a rod to the lever, and may be moved from either side of the bulkhead to which it is fitted. Thus the door may be opened or closed from either side of the bulkhead. The ram, at its lower end, runs through a U leather, and the closing main is connected to the space beneath it.

When the officer on the bridge moves over the pedestal handle, and thus opens the operating valve, pressure flows from the pressure main into the closing main, and thence to the under side of the ram, which is consequently forced over. The slide valve thus uncovers the closing port, and admits pressure to the closing end of the cylinder. To open the door

in such a case, it follows that the pressure on the ram must be relieved. This is accomplished in the following way: Inside the hollow ram is fitted a small miter valve, which is held on to its seat by the pressure in the closing main. A pilot spindle runs through the center of the ram, and terminates at one end against the miter valve, and at the other against the lever. Suitable packing against the spindle keeps it pressure tight. When the lever is moved towards the "open" position, it first depresses this spindle, which lifts the small miter valve off its seat and allows the pressure from the closing main to flow into the tubular ram, past the pilot spindle, which is fluted, and into the exhaust main by suitable ports in the ram and casing. The pressure on its lower end being now relieved, the ram can be operated, as before stated, to open the door. As soon as the handle or lever is released, the pressure in the closing main forces the miter valve back on to its seat, and moves the ram back to the "closed" position, and the door is again closed. Thus any man shut in a compartment may escape, and when he is through, the door will immediately close behind him. Warning of the closing of the doors is given by bells at each door ringing continuously as the door is moving.

In conjunction with the pumps and the pressure main is fitted a governor, on the spindle of which the pressure acts. Any increase over the working pressure of 700 pounds per square inch throttles the steam passing to the pumps. The pressure is thus kept to its normal height, and the pumps are ready at any moment to deliver up to their full capacity.

A circulating valve is also fitted in the pressure main, and allows a slight flow of water to pass into the exhaust main, and back to the tank. This device keeps the temperature of the water throughout the mains constant, thus preventing injury to the pipes, &c.

As it is desirable that the officer on the bridge should know the position of each door, whether open or closed, an electrical indicator is provided; this contains a fascia plate, on which a section of the vessel is engraved. Ruby discs are let into the

plate at different points, and are numbered to correspond with the doors they represent, and these are automatically lighted when the door is open.

THE PROPELLING MACHINERY.

In view of the immense power requisite to propel the new steamers at the designed speed, the Cunard Company, in September, 1903, called to their aid a committee composed of experienced and eminent engineers, to carefully consider the whole question of machinery design. After several months of deliberation and painstaking investigation this committee, in March, 1904, finally reported in favor of the adoption of turbine machinery of the Parsons type.

The steam-producing plant includes twenty-three double-ended and two single-ended boilers, arranged in four compartments. The main propelling machinery consists of two high-pressure ahead, two low-pressure ahead, and two astern turbines. The turbines are of the Parsons type, embodying the latest experience of the Hon. C. A. Parsons and the builders.

Owing to the immense size of these turbines, and in order to comply with the Admiralty's requirements as to subdivision, the main propelling and auxiliary machinery are located in nine different watertight compartments. In the

Table VIII.—LIST OF AUXILIARY ENGINES IN THE BOILER AND MACHINERY COMPARTMENTS.

In High and Low-Pressure Engine Rooms.

Feed pumps..................	Six pairs of Weir's..	18 in. and 13½ in. by 30 in.
Hotwell pumps.............	Four Weir's..........	12½ in. and 14½ in. by 30 in.
Surface heaters.	Two Weir's............	1,750 sq. ft. each.
Contact heaters............	Ditto	48 in. in dia. casing.
Feed-water filters.........	Two Harris's.........	36 in.
Ditto, auxiliary............	Ditto	20 in.
Aux. circulating pumps..	Two Allen's 10-in...	36-in. disc engine, 7 in. by 6 in.
Auxiliary condensers.....	Two Clydebank......	2,000 sq. ft. each.

Bilge pumps..................	Four Weir's..........	8 in. and 10 in by 21 in.
Fresh and condensed-water pumps	Two Carruther's.....	Two 6 in., two 6 in. by 6 in.
Water service circulating pumps	Ditto	Two 7½ in., two 10 in. by 12 in.
Sanitary pumps............	Two Weir's............	10 in. and 10 in. by 10 in.
Auxiliary air pumps......	Ditto	10 in. and 22 in. by 12 in.
Oil pumps...................	Six Weir's.	7 in. and 8½ in. by 15 in.
Gland and jacket drain-tank pump	One Carruther's duplex	Two 4½ in. and two 5 in. by 5 in.
Sluice-valve engines......	Two Clydebank......	Two 6 in. by 6 in.
Reversing engines.........	Four Brown's.........	6 in by 8 in.

In Pump Rooms.

Circulating pumps.........	Two sets Allen's.....	Each set with two 18-in. by 10-in. engines and two 32-in. discharges, and two 42-in. discs.
Wet-air pumps..............	Four twin Weir's....	14 in. and 40 in. by 24 in.
Dry-air pumps..............	Ditto	7 in. and 24 in. by 7 in.
Aft water-service pump..	One Carruthers' duplex	Two 6 in., two 7 in. by 7 in.

Evaporator Rooms.

Wash-deck and fire pumps	One Weir's............	10 in. and 10 in. by 10 in.
Evaporators..................	Six Liverpool Engineering Co.'s	
Distillers......................	Four Ditto.	

Pumps for evaporators: Two circulating pumps, two feed pumps and two brine pumps.

In Boiler Rooms.

Assistant feed and ash-ejector pumps	Four Weir's..........	14 in. and 10 in. by 14 in.
Ballast pumps...............	Two Carruthers' duplex	Two 8 in., two 10 in. by 10 in.
Ash ejectors.................	Eight sets Mechan.	
Ash hoists.....................	Eight sets Crompton.	

Forced-draft fans and motors: Eight sets of fans and motors, each set with four fans and two motors; fans, 66 in. in diameter, Allen's.

Turbo-dynamo..	Four Parsons'........	375 kilowatts.

largest and most central of these are located the two low-pressure and the two astern turbines, the whole of the feed pumps, hotwell pumps, oil pumps, and the pumps for the Stone-Lloyd hydraulic system of closing bulkhead doors. In each of the two wing compartments, separated from this central compartment by a longitudinal bulkhead, are placed respectively the high-pressure turbines, the auxiliary condensers, auxiliary circulating and air pumps, also the water-service, fire and bilge and fresh-water pumps. Aft of these, in separate wing compartments, are placed the evaporating and distilling plants, underneath which runs a portion of the high-pressure tunnel shafting. In the fifth compartment, extending right across the ship, are placed the four main condensers, and underneath these the feed tank. Aft of these again are two auxiliary-machinery rooms, each two-storied and each self-contained. In the lower are placed the main wet-air pumps, dry-air pumps and main centrifugal circulating pumps; and in each of the upper flats the electric generating stations, in which the power is supplied by two Parsons turbo-generators. Abaft these are the shaft tunnels.

The starting gear is situated on an upper platform at the forward end of the main engine room, and from this position practically everything can be controlled. The main regulating-valves for the high-pressure and the maneuvering turbines are arranged and worked very similarly to those on the *Carmania*, the Brown's engines being located on the platform, so that, in the event of any hitch, the valves can be worked at once by hand in place of steam power.

STEAM-GENERATING PLANT.

Although the turbines naturally awaken greater interest than the steam-generating plant, it will be more consistent with the natural order if, in entering upon a detailed description of the machinery, we deal first with the boilers.

There are, as already stated, twenty-three double-ended and two single-ended boilers in the ship, situated in four separate water-tight compartments. The forward boiler room, desig-

nated No. 1, has two single-ended and five double-ended boilers, and in each of Nos. 2, 3 and 4 boiler rooms there are located six double-ended boilers, in groups of three athwart the ship. In Nos. 2, 3 and 4 boiler rooms the bunkers are situated on the sides of the boilers and boiler casings; but in No. 1 boiler room, on account of the increasing fineness of the ship, recourse had also to be had to a large athwartship bunker, forward of which is a hold available either as a reserve bunker or for general cargo.

The double-ended boilers, are 17½ feet in diameter by 22 feet long. In each of these boilers there are 344 stay tubes and 720 plain tubes, the total being 1,064. The total heating surface is 6,593 square feet and the grate area 168.65 square feet. There are separate combustion chambers to each furnace, and the water and steam spaces are exceedingly ample. The single-ended boilers differ from the others only in their length.

The furnaces have a collective grate area of 4,048 square feet, while the total heating surface is 158,350 square feet. For each group of boilers there is a separate funnel, all being of the same dimensions, as the heating and grate surfaces of each boiler room are identical. These funnels are elliptical in form, the extreme dimensions of the outer funnel bing 19 feet by 26 feet, while the height above the grate level is 130 feet. We may add that the shells of the boilers are made of high-tensile steel, of a maximum tensile strength of 36 tons to the square inch; but the fronts and backs, as also all rivets, including those for the shells, are made of ordinary mild steel. The 192 furnaces are of the cambered type, which is a specialty of the Atlas Works, Sheffield, of Messrs. John Brown & Co., limited. These furnaces were made to a diameter over ribs of 4 feet and $1\frac{9}{16}$ inches, and for a working pressure of 195 pounds per square inch. The front and back ends of these furnaces were finished off in accordance with the latest practice, having their landings turned so as to ensure a perfect fit.

There are four main leads of steam piping, one from each boiler room and all four are carried right aft to the forward engine-room bulkhead, on the forward side of which a stop

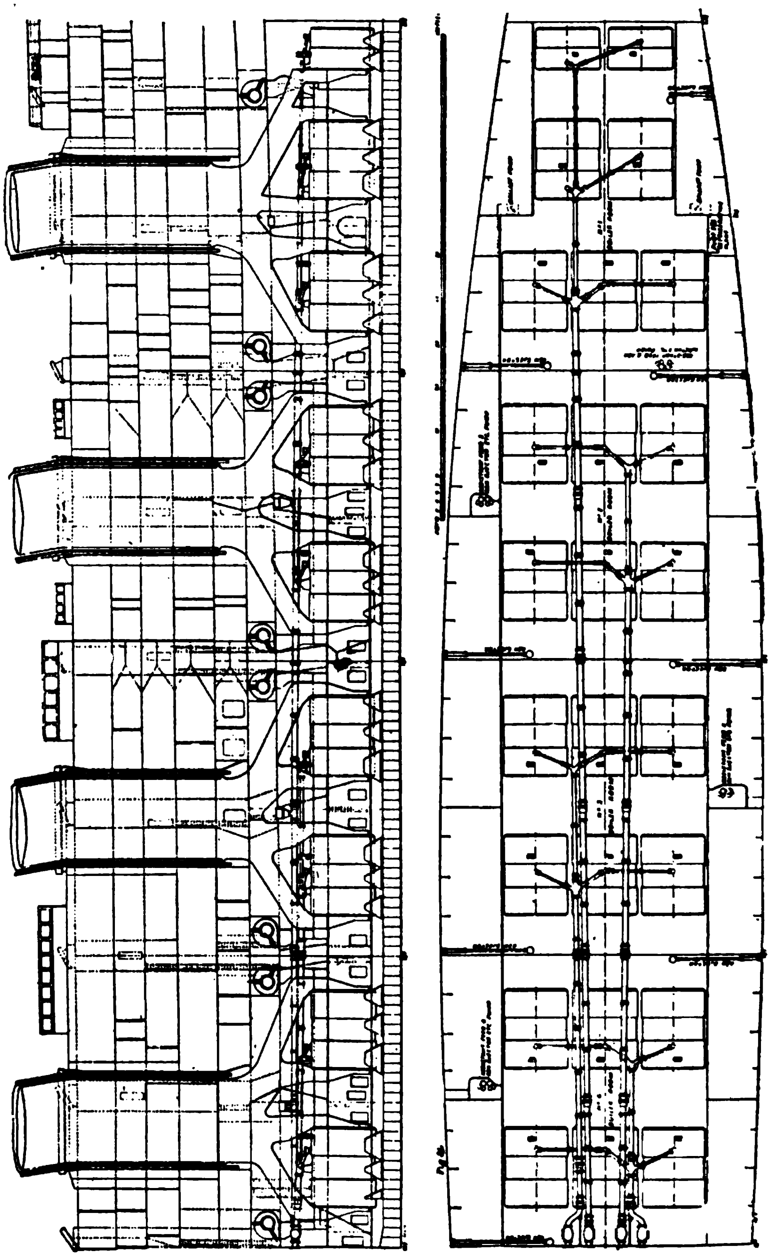
valve is fitted to each of the four lines, which can be worked both from the engine room and from the boat deck.

THE FORCED-DRAFT FANS.

The forced draft is on the well-known Howden system of heated air and closed ash pits, the forced-draft fans being driven by electric motors.

There are thirty-two motor-driven fans, of Messrs. W. H. Allen, Son & Co.'s manufacture, arranged in pairs, and driven by sixteen electric motors. The fans are throughout of sheet steel, and have impellers of the single-inlet type, 66 inches in diameter, and are capable of producing an air pressure of 3½ inches when running at 450 revolutions per minute. The motors are capable of developing 50 brake horsepower, receiving curent at 100 volts, when running at 450 revolutions per minute. These motors, also of Messrs. Allen's manufacture, are completely enclosed, but inspection doors are provided at the commutator end of each motor, which can be readily removed. As the sets have to run in an atmosphere at high temperature, special means have been provided for the ventilation of the motors. The fan next to the commutator end of the motor in each set is provided with a small auxiliary impeller and separate casing, for the sole purpose of supplying air for the ventilation of the motor. This casing is connected so as to deliver the air into the bottom of the commutator casing, the outlet being at the opposite end of the motor casing. The controllers for these motors are also of Messrs. Allen's manufacture, and are arranged to give a large variation of speed, rising in equal increments from about 185 to 500 revolutions per minute. The fans are also provided with water-pressure gauges connected to the fan casing, and with tachometers.

The twin-screw ship *Vera,* built at Clydebank in 1897, was the first instance in which steam-driven fans were dispensed with for forced draft in boiler rooms. The resulting freedom from vibration and noise, and the ready means of regulation from the starting platform, proved so satisfactory that



GENERAL ARRANGEMENT OF BOILERS OF THE "LUSITANIA."

motor-driven fans were subsequently installed, with equally good results, in the sister ship *Alberta,* built at Clydebank for the London and South-Western Railway Company, and the Steamship *Antrim,* built by John Brown & Company in 1905 for the Midland Railway Company. In these ships the controllers were fitted in the engine rooms and the fans entirely regulated from that position; but in the case of the *Lusitania* the distance from the engine room to the boiler rooms is so great that, while it was desired to be able to regulate the fans from the engine room, it was at the same time also considered advisable to retain the ordinary controllers attached direct to the fans in the boiler rooms, and this required result has been most satisfactorily attained by the very ingenious apparatus of Messrs. Siemens Brothers.

When it is desired to alter the position of any one of the fan controllers the operator works the switch handle of the sender to either "fast" or "slow," the handle, when released, being returned to the central position by a spring. This switch causes the motor on the controller to run either forward or backward, and through the gearing to turn the controller barrel either forward or backward. Geared to the controller barrel is the indicator switch, the indicator being situated just above the sender switch, and consisting of a revolving drum, on the periphery of which are figures and letters corresponding with those on the top of the controller case; this drum has an electro-magnetic control, worked from the indicator switch geared to the controller barrel, so that the figures or letters on the drum, as seen through the window of the indicator, will correspond with the position of the controller barrel. In order to ensure great accuracy of position of the controller barrel, a bell is placed in the indicator which is connected to a contact ring in the indicator switch, having portions of the ring cut away, so that the bell will continue to ring as long as the controller barrel is not in its exact position. It will thus be seen that the operator, when watching the figures on his indicator, can see exactly what is taking place on the controller, the bell telling him whether the controller is in its exact position or not.

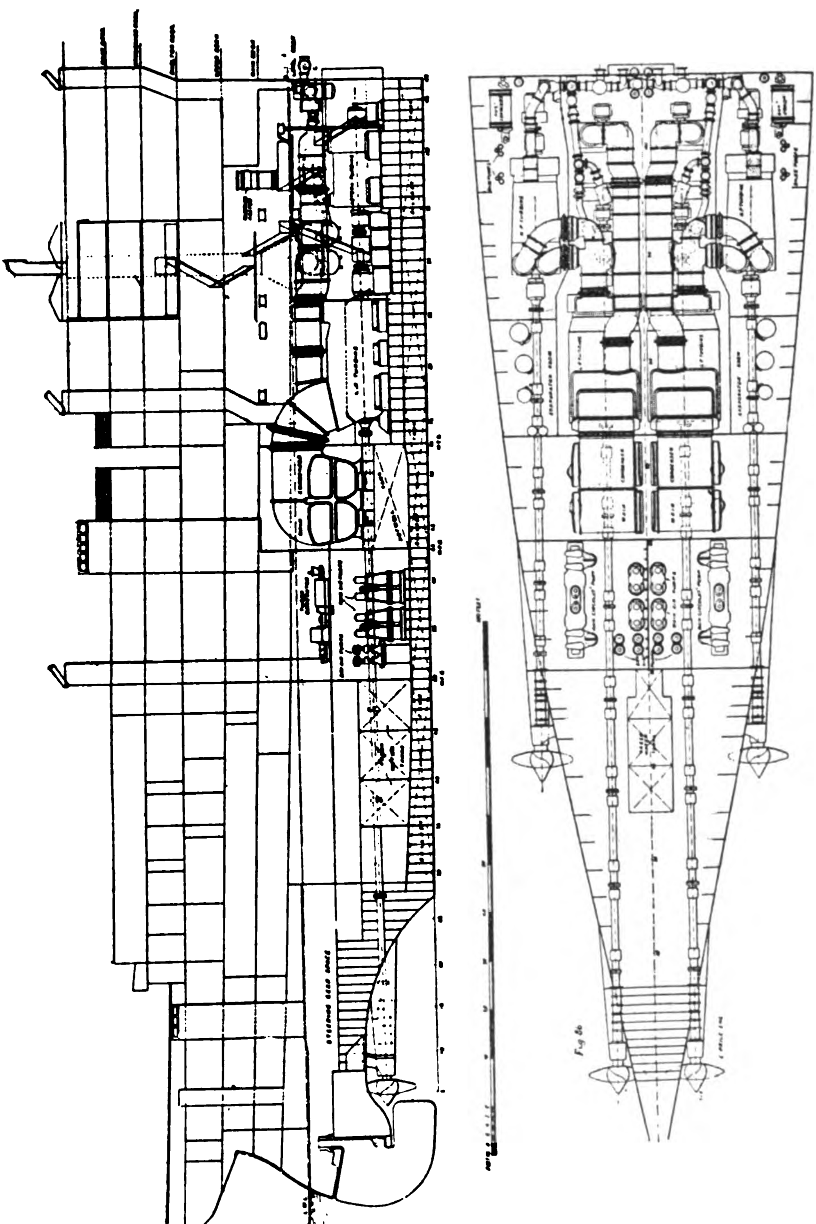
The working of the indicator is positive, not step by step, and the instrument is of the same type as is used in the Navy for controlling the firing operations. The motor on the controller is geared to the controller barrel by means of a slipping clutch, so that if the operator runs the motor beyond what is necessary to turn the barrel, no harm will be done. The whole apparatus is watertight. An ammeter is fitted above the indicator to record the amount of current used by the fan.

The air inlets to the fans have been arranged as trunks extending up to the boat deck, and, instead of the usual array of cowls, which are always unsightly, especially where, as in this ship, they require to be of great size, they are circular shafts fitted on top with hinged covers. The extent of the opening can be varied, and gear is also arranged so that the cover may be rotated to suit the wind when on the beam. When the vessel is steaming ahead the opening will be, as a rule, towards the bow.

THE DISPOSAL OF ASHES.

One of the difficulties which have to be dealt with in vessels burning such a large amount of coal as will be the case in this instance, is the disposal of the ashes. As a matter of fact, the expeditious disposal of the ashes has a direct effect on the steaming capabilities of any high-powered vessel, as it is impossible foı the firemen to give proper attention to the firing when the stokeholds are hampered with ashes. Eight See's ash ejectors are fitted to the *Lusitania* (and also in the sister vessel), and with this apparatus all that is necessary is for the firemen to shovel the ashes into a hopper in the stokehold, after which they are dealt with by the apparatus without further manual labor on the part of the firemen, and are discharged twenty or more feet clear of the ship's side. Ten to fifteen minutes each watch will suffice to clear each stokehold of ashes.

An alternative system of dealing with the ashes, and for disposing of them when in harbor, has been fitted by Messrs. T. Albert Crompton & Co., London. This system is known as the Crompton atmospheric silent ash hoist.



GENERAL ARRANGEMENT OF TURBINES OF THE "LUSITANIA."

THE TURBINE INSTALLATION.

Having dealt with the boilers and their incidental machinery, we follow the steam to the turbines, which are the most interesting feature in the ship. The high-pressure turbine rotor drums are 96 inches in diameter, the astern drums 104 inches, and the low-pressure drums 140 inches.

For such immense turbines it will naturally be inferred that the forgings, etc., required in their construction would also be of very large size. The steel forgings for the turbine drums, rotor spindles and straight shafting were manufactured at Messrs. John Brown & Co.'s Atlas Works, Sheffield. The turbine drums were all hollow forged, and the low-pressure ahead drums are the largest hollow forgings that have been made up to the present time. The sizes of these drums, as delivered to Clydebank, were as follows: Outside diameter, 11 feet, 8⅝ inches; inside diameter, 11 feet 4 inches; length, 8 feet 2 inches; with metal 3 inches thick. To manufacture these enormous forgings necessitated a good deal of scheming and ingenuity on the part of the staff at the Atlas Works. It may be of interest to give a few data of the forgings for these particular drums. The ingots used were 60 inches in diameter across the flats, and weighed 42 tons each. These were forged down to 54 inches in diameter, and an 18-inch hole trepanned through, making hollow pieces weighing about 27 tons each, which were subsequently expanded into drums 11 feet 10½ inches outside diameter. These forgings were then rough turned and bored in a lathe that had been specially altered for the purpose. These drums were so large that it was impossible for the railway companies to carry them, and it was necessary to take them by road from Sheffield to Manchester, where they were shipped direct to Glasgow.

The whole of the rotor spindles and shafting were also manufactured at the Atlas Works, all being, of course, hydraulically forged. The rotor spindles are extremely large, the majority of them having coupling flanges 43 inches in diameter, and some being as large as 40 inches in diameter in the body. The intermediate shafting is 20 inches in diameter, and the tail

shafts 22⅛ inches in diameter by over 30 feet long. The rotor spindles, thrust shafts and tail shafts are all made of high-tensile steel.

The throttle valves, valve covers, strainers, expansion joints, etc., are also of very large proportions, and were made as steel castings by the Robert process, which is also one of the specialties of the Atlas Works. The uniformity and superiority of the physical results obtainable by this process are particularly striking, and are attained with less difficulty than by any other method at present in use.

The cast-steel dished wheels on which these drums are shrunk were supplied by Thomas Firth & Sons, Limited, Sheffield, who have made this particular type of turbine castings a speciality, and have supplied them to practically every firm of turbine builders in Great Britain. Each wheel weighed 11¾ tons. In all, Messrs. Firth supplied about 440 tons of castings. The whole of the castings were subjected to tests and inspection of the Board of Trade and Lloyds. Owing to the great contraction that takes place in steel—double that of cast iron—the greatest possible care has to be exercised in the molding of such huge wheels to avoid possible failures, and only long foundry experience and skill can overcome the difficulties.

The turbine blades vary from 2¼ inches to 22 inches long. In the longer blades the necessary radial and lateral stiffness is obtained by means of three rows of shrouding, in which expansion is allowed for. The longer blades of the low-pressure turbines are bound together by two circumferential strips laced with copper wire and soldered; and, in order to prevent distortion, due to the differences of expansion of the drum and the brass strip, very ingenious expansion joints had been devised. The binding strip was divided into short lengths, connected by brass tubes, in which they could slide.

The turbine spindle steam glands, valves, governors and system of lubrication which have answered so admirably in the *Carmania,* are again adopted in the new vessel.

The lifting-gear for the low-pressure turbine has to be capa-

ble of lifting the immense weight of 115 tons; and the provision of suitable apparatus and means of stowing the immense receiver and exhaust pipes was one of the many problems which have been so successfully solved by the builders of this vessel.

THE CONDENSERS.

The steam having performed its work in the turbine, the next step is to return it in the form of feed water to the boilers. In a leviathan such as the *Lusitania* the sequence of steps is practically similar to that in any ordinary installation, there being little scope of novelty in method. The number, size and arrangement of the auxiliaries form therefore the chief features which call for note.

There are four main condensers, arranged in pairs, each unit containing 20,700 square feet of cooling surface, giving an aggregate of 82,800 square feet. A 32-inch bore circulating-water pipe is led to each condenser from the large centrifugal pumps. The two auxiliary condensers, which are situated at the forward end of the high-pressure engine room, have a collective cooling surface of 4,000 square feet, and have separate circulating and air pumps.

Each of the four main condensers is fitted with the Harris-Anderson patent condenser-tube protector, supplied by the Harris Patent Feed-Water Filter, Limited, for preventing corrosion of the tubes, ferrules, etc. The principle of the system is the introduction into the circulating water of a metal which is electro-positive to the metal forming the tubes of the condenser, the tubes and the electro-positive metal being connected together. The tubes are saved at the expense of the metal that is connected to them. The composition of this protective metal may be altered to suit particular cases. The metal usually employed, however, is electro-positive to nearly all the various alloys of copper and zinc, and it, moreover, retains its protective properties till it is entirely dissolved. There is found to be no trouble with insoluble deposits on its surface. Though not always convenient, the apparatus is sometimes fitted inside the end of the condenser. The makers prefer, however, to

provide an independent vessel as a container for the protector, which vessel communicates with the water space by two pipes, which can be closed by valves when desired, thus enabling the protector to be inspected or renewed without interfering with the working of the condenser.

Each tube in the condensers of the *Lusitania* is brought into meallic contact with the tube plate by means of a soft-metal washer inserted on the top of the packing in the stuffing box. When the ferrule is screwed up the washer spreads out, thus forming the necessary contact between the tube and tube plates. Protective metal blocks are secured in the water ends of the condensers, some in direct contact with the tube plate, and others connected by cables to terminals at the opposite end to the blocks. Any corrosive action likely to attack the tubes is thus transferred to the protector, and pitting of the tubes, ferrules, etc., is avoided.

THE CIRCULATING PUMPS.

The circulating pumps, by Messrs. W. H. Allen, Son & Co., Ltd., Bedford, present some novel features. The main circulating engines consist of eight "Conqueror" type centrifugal pumps, having suction and discharge branches 22 inches in diameter, and arranged in four pairs, the discharge branches from each pair uniting into one common discharge of 32-inch diameter. Each pair of pumps is driven by a single-cylinder high-speed forced-lubrication engine of Messrs. Allen's well-known standard type, the engines again being arranged in pairs. Thus the main pumping machinery is grouped into two sets. The engine shafts can be coupled together in pairs, the engines running as two pairs of two-cylinder high-pressure engines, an arrangement of weights having been provided whereby the balance is exceedingly good under these conditions. The steam distribution is effected by means of piston valves, the valve chest being cast in each case in one piece with the cylinder, the whole being of exceedingly close-grained and tough metal. The cylinders have a diameter of 18 inches, with a stroke of 10

inches and together are capable of developing 350 brake horse-power at a speed of 300 revolutions per minute, receiving steam at 160 pounds per square inch, and working against a back pressure of 10 pounds per square inch. The cylinder bodies and covers are well lagged with silicate cotton, and neatly covered with burnished sheet brass, and fitted with the usual drain-cocks and relief valves, presenting a very smart appearance. Cast in one with each cylinder is a substantial cast-iron distance piece, which is faced square with the bore of the cylinder for bolting to the top of the engine trunk. This distance piece is provided with openings through which access can be obtained to the stuffing boxes, which are all packed with United States metallic packing.

As stated above, the cylinders are arranged in pairs, each pair standing upon a cast-iron trunk of very rigid design, which carries the slide faces for the crossheads: these faces are accurately scraped up and finished square with the top and bottom faces of the trunk. In front are three doors which can be readilv removed for inspection and adjustment of the working parts. Special oil and water glands are fitted to the top of the trunk where the piston and valve rods pass through it. This effectually prevents the oil from working up to the cylinders from the crank chamber, and precludes water from the cylinders entering the crank chamber. The whole of the trunks and cylinders complete stand upon a rigid box-section bed plate, in which is arranged the oil reservoirs, filters and oil pumps, each of these latter fittings being arranged in duplicate, so that the engines may be disconnected from each other and run separately. The oil pump is of the valveless oscillating type, and is driven from the engine eccentric, and delivers oil under pressure to all the working parts. Each pump is also provided with an oil-pressure regulator, whereby the pressure can be regulated while running.

On the front of the engine trunk are arranged the oil and steam pressure gauges, in close proximity to the stop valves; and neat transmission gear is arranged for operating the drain cocks of the engine.

Each end of the bed plate is provided with an extension for

bolting to a similar extension of the gun-metal pump casing. An outer bearing is also provided between the flywheel of the engine and the pump. All the main bearings of the engine and pumps are of cast steel, lined with white metal, with the exception of the crosshead bearings, which are of gun metal. Separate barring gear is also provided for each flywheel. Owing to the engines being of the totally-enclosed type, tachometers are also provided for each set to continuously indicate the speed of the machinery.

As stated above, the main pumps are constructed throughout, both casings and impellers, of gun metal, the casings being 7-16 inch thick, and the discs having a diameter of 42 inches. The pump shaft is of forged bronze, and carried in bearings external to the pump. The shaft enters the pump casing at each end through stuffing boxes having gun-metal glands and special provision for lubrication.

In addition to the above main circulating pumps and engines two auxiliary circulating pumping engines have been fitted, each pump being of gun metal, and having suction and discharge branches 10 inches in diameter, while the diameter of the impeller is 36 inches. The engine is of Messrs. Allen's standard open type, having a single cylinder 7 inches in diameter, with a stroke of 6 inches. The steam distribution is effected by means of a piston valve. The cylinder, valve chest and cover are lagged with silicate cotton, and neatly covered with blue sheet steel. The engine-bed plate is rigidly connected to the casing of the pump, and external pump bearings are provided, being lined with white metal. The piston-rod and valve-rod glands are also fitted with metallic packing of the United States type, and lubrication is provided from a central oil box, from which oil is carried by pipes to the various bearings. The cylinder is fitted with the usual grease cup and spring relief valves and drains.

FEED-WATER PUMPS AND HEATERS.

To the condensers, four in number, are connected four Weir wet-air pumps, 40 inches in diameter by 24 inches stroke. These are of Messrs. G. and J. Weir's twin type, having two

steam cylinders, two pump barrels, with the pump rods cross connected by a beam. Steam is admitted to both cylinders by a single valve of the Weir pattern, designed specially for air-pump duty, but comprising the usual and distinguishing features of the well-known Weir valve. Gun metal has been adopted for the pump barrels, the buckets, foot and head valve seats, which latter are fitted with Kinghorn valves and gun-metal guards. The cylinders are supported on a cast-iron entablature set on angle wrought-iron columns. The piston rods are of steel, connected by a crosshead with the pump rods, which are of manganese-bronze, and work in vertical guides.

In addition to these wet-air pumps, which are capable of maintaining the requisite vacuum when the system is reasonably tight, provision is made for unexpected or accidental leakage by fitting four sets of Weir double dry-air pumps, 24 inches in diameter and 7 inches stroke, for dealing with air only. In these the air-pump chambers are situated over the steam cylinders of a double-connected enclosed high-speed engine. These chambers are of gun metal, and are of the single-acting type. The air passes into the barrel above the buckets through annular openings, and is forced through the head valves on the up-stroke of the pump. The compression of the air results in a certain rise of temperature, which is taken care of by a small supply of circulating water, which passes through the chamber and carries off the heat. Steam is admitted to the engine by a piston valve controlled by a governor fitted on the shaft in the usual manner.

From the air pumps the feed water passes to the hotwell, from which it is taken by four Weir hotwell pumps 14½ inches by 30 inches, of the firm's light-duty type, fitted with Kinghorn valves, and having gun-metal liners, brackets, and manganese-bronze rods. These pumps are automatically controlled by Weir control gear fitted in the hotwell, so that the speed of the pumps corresponds to the quantity of water passing into the chamber. The feed water is discharged by these pumps through two Weir surface feed heaters, where the ex-

haust steam from all the auxiliaries (with the exception of the turbo-generators) is utilized to heat the feed, and as this steam is impregnated with oil, it flows, after condensation, by gravity through an oil filter into the hotwell tank. In addition to this feed heater there are also fitted two Weir direct-contact heaters, into which the exhaust steam from the turbo-generators is led. There is here also control gear for regulating the speed of the main feed pumps. These consist of three pairs of Weir standard feed pumps, 13½ inches in diameter, with a 30-inch stroke, which are supplemented by a duplicate installation of auxiliary feed pumps of the same size and number. These pumps have all gun-metal barrels, with manganese-bronze valves and pump rods, steel piston rods, with the requisite suction and discharge stop valves for drawing from the feed heaters and discharging to the boilers.

In addition to these auxiliaries Messrs. G. & J. Weir, Limited, have also supplied four duplex pumps of special design for ash-ejector and auxiliary feed duty, 10 inches in diameter, with a 14-inch stroke, three duplex pumps for sanitary and wash-deck purposes, also four single direct-acting bilge-pumps, 10 inches in diameter, with a 21-inch stroke. For the supply of oil to the turbine bearings, six of their special direct-acting lubricating pumps are fitted. For dealing with the water and air from the auxiliary condensers, they have furnished two of their latest type of single direct-acting air pumps, known as the "Monotype" pattern, 22 inches in diameter, with a 12-inch stroke. These represent the latest developments in air-pump design. The installation of Weir auxiliaries is very complete and representative, and practically handles the feed water from the time it leaves the condenesr until it is returned to the boilers; a responsible duty which calls for most reliable equipment.

FEED-WATER FILTERS.

Two feed-water filters of the well-known Harris type, supplied by the Harris Patent Feed-Water Filter, Limited, are fitted in connection with the hotwell pumps of the main tur-

bines, and filter the water on its passage to the feed heaters. The filters are each 36 inches in diameter, and of gun metal throughout, the principal feature in their internal construction being the central sludge outlet—an ingenious arrangement, by which the filtering area is divided into eight separate sections, each of which can be sludged out independently of the rest, the whole force of the reversed current of the water when cleaning, being concentrated on only one-eighth of the surface, so that the cleaning is most efficient, and can be effected in a few minutes without the necessity of opening up. The filters present a most compact appearance, and everything is well arranged to facilitate their ready manipulation. Two smaller filters, 20 inches in internal diameter, also in gun metal, are fitted in connection with the auxiliary machinery. These filters are of the same type as the larger ones, but with all valves self-contained.

DISTILLERS AND EVAPORATORS FOR MAKING-UP THE FEED WATER.

The distilling machinery is of Quiggin's well-known type, and was manufactured by the Liverpool Engineering and Condenser Company, Limited, Brunswick Dock, Liverpool.

There are two complete sets of plant in the *Lusitania,* and these supply the whole of the distilled water required for all purposes, the total capacities of each plant being, for cooking and drinking purposes, 18,000 gallons per 24 hours; for baths and washing, 15,000 gallons per 24 hours; while the evaporators for feed make-up purposes for the boilers, when working compound effect, supply 240 tons per 24 hours, and when working single high-pressure effect, 350 tons per 24 hours.

Each plant consists of one evaporator for the production of distilled water, and two for feed-make-up purposes, the two latter being arranged to work in series—compound, or separately—single effect. All the evaporator shells are made of rolled naval brass, double riveted. This is for the purpose of reducing the weight as far as possible. The ends of the evaporators, all the mountings, as well as the frame and doors, are

constructed of gun metal. The coils are made so that they can be withdrawn bodily, and are all interchangeable, and the coils can be taken out separately for the purpose of cleaning and for inspection; while in order to facilitate this operation a complete spare heating surface is provided for each size of evaporator, in order that the coils may be replaced by a clean set, when required, in a few minutes. An automatic feed-water regulator is provided for each evaporator, and this maintains the water level in the evaporator at a constant height.

The condensers have coils of solid-drawn copper, and are tinned inside and outside; the coils can be withdrawn bodily with the cover by simply unscrewing a nut on the spigot end at the bottom connection to the filter. The sectional area of the coils diminishes from top to bottom, but each coil has a parallel surface throughout. The inlet for the steam is of full bore where the steam enters, but is gradually reduced in area to a crescent section, until at the outlet end it is only about one-third of the original sectional area. The volume of the steam is reduced as it condenses, and is kept in contact with the condensing surface, owing to the diminishing area of the coil. It is claimed that in this way the surface is rendered much more effective than it would be if the coils were of the same sectional area throughout. The filter, which is charged with animal charcoal and limestone chips, is in the base of the condenser. As a means of aerating the distilled water there is a pipe fitted, which is tapped from an iron-pipe connection, and there is a door for access to the filter. The circulating water enters and flows as shown. There are two condensers in each set. The shells of the condensers are of galvanized mild steel.

In each set of apparatus there are three pumps, namely, one vertical duplex circulating pump, one vertical duplex evaporator feed pump, and one vertical single direct-acting type brine pump for pumping the brine from the low-pressure evaporator (when working compound effect); after the water has been diluted and cooled with sea water it is pumped overboard. All these pumps are made with solid gun-metal water ends.

PUMPS FOR SUNDRY DUTIES.

In the engine room there are a great variety of pumps for sundry duties. Many of these have been supplied by Messrs. J. H. Carruthers & Co., Limited, Polmadie, Glasgow. The arrangement of framing in this type gives very free access to all the moving parts. All the important joints of the valve gear are adjustable. The water valves are easily examined through the front doors of the pump.

Among the pumps supplied are the following: Two for ballast service, with cylinders 8 inches and 10 inches by 10 inches; two for water service, with cylinders 7½ inches and 10 inches by 12 inches; two for washing decks 6 inches and 6 inches by 6 inches; one for sanitary service 6 inches and 7 inches by 7 inches. All of the pumps have gun-metal ends.

VENTILATION OF THE ENGINE ROOM.

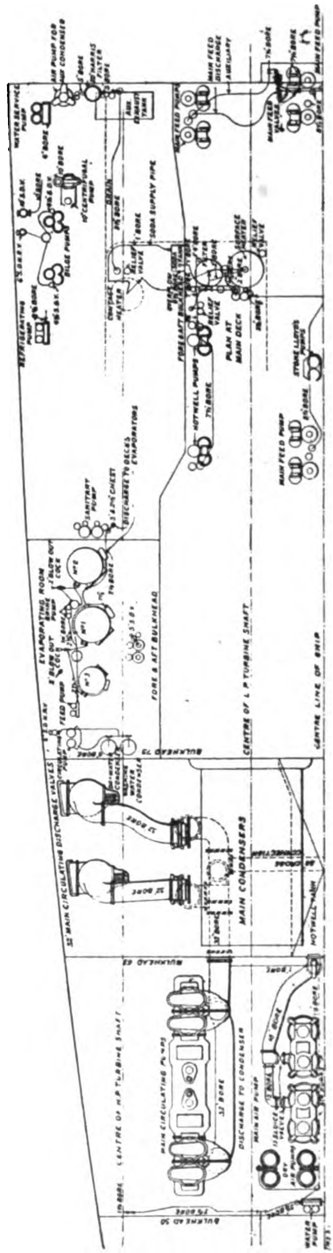
Messrs. Laurence, Scott & Co., Limited, Norwich, supplied twelve fans of 35 inches, two of 30 inches and two of 25 inches diameter, all electrically driven and adapted for the ventilation of the engine room. The outputs specified were respectively 26,000 and 14,000 cubic feet per minute, with free discharge at 315 and 450 revolutions per minute, the fans being direct driven and carried on an extension of the motor spindle. The company's standard type of semi-enclosed motor was adopted, fitted with gauze grids, the magnets being series wound for the reasons given below. In view of the high temperature of the situations in which some of these fans work, the motors were made large, and the temperature rise in a six hours' run was kept below 50 degrees F. The armature is all built up on a cast-iron quill, and is self-contained and independent of the shaft, on to which it is slipped when completed. Series winding was adopted for the magnets, as this gives better regulation of the load than shunt winding would do. The power required by a centrifugal fan at a constant speed goes up rapidly as the resistance to its free discharge is removed, reaching a maximum when disconnected altogether from its air trunks. The variation in speed of a series-wound

motor tends to correct the effect of variations in the resistance to discharge of the air, and keeps the load on the motor and the volume of air more nearly constant than would be the case if a shunt motor were used. The series winding also gives a simple method of speed control without the use of resistances. For slow speed all four field coils are arranged in series with each other and the armature. For full speed the field coils are arranged in two parallel circuits, each of two coils in series, these being still in series with the armature. The motor is then running with a lower resistance in series with the armature and with a weaker field, and therefore at a higher speed. The barrel controller is protected by an overload and no-voltage device. In the event either of an overload or failure of supply, the barrel carrying the contracts flies to the "off" position, even if the operating handle is being held "on." The fans are Messrs. Davidson & Co.'s make, of the well-known Sirocco type, and, like the motors, are amply large for the work.

ARRANGEMENT OF AUXILIARY MACHINERY.

The condensers are in a separate compartment abaft the main central engine room and over the two shafts driven by the low-pressure turbines. The circulating pumps and the air pumps are in a compartment abaft the condensers. The air pumps are accommodated between the shafts, and are at a lower level than the circulating pumps.

The evaporator and distilling plant are in a compartment in the wings of the ship, over the outer shafts and abaft the high-pressure turbine. In this way space is admirably economized. It will be understood by those who have studied the preceding figures that the high-pressure turbines in the wing compartments are not in the same athwartship line as the low-pressure turbines, the former being considerably in advance. This disposition of the turbines has enabled the larger of the auxiliary engines to be grouped in the forward part of the central engine compartment in the vacant space around the astern turbines, which are at the forward end of this center engine compartment. The main feed pumps occupy a central



GENERAL ARRANGEMENT OF AUXILIARY MACHINERY AND DISCHARGE PIPE.

position, and the hotwell pumps are in the wing; while to the forward of them, but on a higher level, are the filters and the feed-water heaters. Still further forward, against the main engine-room bulkhead, are the main feed pumps and valves.

SOME RECENT RECORD PERFORMANCES.

Record-breaking steamers.	Time.			Speed.	Best day's run.
	days.	*hrs.*	*min.*	*knots.*	
In 1840 *Britannia's* trip—Liverpool to New York	14	0	0	$8\frac{1}{2}$	...
In 1862 *Scotia's* trip—Liverpool to New York.	8	22	0	13	...
Servia, 1884 ... Outwards	7	10	47	...	...
Homewards	6	23	57	...	...
Oregon, 1884 ... Outwards	6	10	9	...	...
Homewards	6	16	59	...	...
America, 1884 ... Homewards	6	14	18	...	...
Umbria, or *Etruria* ... Outwards	6	1	44	19.3	501
Homewards	6	3	12	19.1	...
Paris, or *New York* ... Outwards	5	14	24	20.7	530
Homewards	5	19	57	20.1	...
Campania, or *Lucania*, 1904 .. Outwards	5	7	23	21.82	562
Homewards	5	8	38	22.01	533
Kaiser Wilhelm der Grosse, 1902, Cherbourg-Sandy Hook	5	15	20	22.81	580
Kaiser Wilhelm der Grosse, 1901, Sandy Hook-Plymouth	5	10	0	23.0	553
Deutschland, 1903 ... Cherbourg-New York	5	11	54	23.15	...
Deutschland, 1900 ... New York-Plymouth	5	7	38	23.51	...
Kronprinz Wilhelm, 1902 ... Cherbourg-Sandy Hook	5	11	57	23.09	581
Kronprinz Wilhelm, 1901 ... Sandy Hook-Plymouth	5	8	18	23.47	561
Kaiser Wilhelm II, 1904 ... Cherbourg-Sandy Hook	5	12	44	23.12	583
Kaiser Wilhelm II, 1906 ... Sandy Hook-Plymouth	5	8	16	23.58	564

The *Deutschland's* westward mean speed of 23.51 knots, made over a long course, and not, therefore, a record in point of time, is equivalent to steaming from Queenstown to Sandy Hook in about 4 days 23 hours; and the *Kaiser Wilhelm II's* homeward mean speed of 23.58 knots would bring her to Queenstown in a few minutes' less time.

DIMENSIONS AND PERFORMANCES OF NOTABLE ATLANTIC STEAMERS.

Steamer's name.	Builders.	Date.	Molded dimensions. Length.	Molded dimensions. Breadth.	Molded dimensions. Depth.	Proportion of length. To beam.	Proportion of length. To depth.	Draught.	Displacement.	Gross tonnage.	Cylinders. Diameter in inches.	Cylinders. Stroke.	Boilers. Heating surface.	Boilers. Grate area.	Boilers. Working pressure.	Indicated horsepower.	Speed on trial.
			ft.	ft. in.	ft. in.			ft. in.	tons.			in.	sq. ft.	sq ft.	lb.		knots.
Great Eastern	Scott Russell	1858	480	83 0	57 6	8.192	11.826	25 6	27,000	24,360	Screw, four 84-in.; paddle, four 74-in.	S., 48 P., 17	...	...	30	7,650	14.5
Britannic	Harland & Wolff	1874	455	45 0	36 0	10.111	12.640	23 6	8,500	5,004	Two 48-in., two 83-in.	60	...	...	70	5,500	16
Arizona	Fairfield Co.	1879	450	45 2	37 6	9.955	12.000	22 0	...	5,147	One 62-in., two 90-in.	66	...	...	90	6,300	17
Servia	Clydebank Works	1881	415	52 0	40 6	9.908	12.716	23 3	9,900	7,392	One 72-in., two 100-in.	78	27,483	1,014	90	10,300	17
Alaska	Fairfield Co.	1881	400	50 0	39 8	10.000	12.607	22 0	...	6,932	One 68-in., two 100-in.	72	...	...	100	10,500	18
City of Rome	Barrow Co.	1881	543	52 0	38 9	10.432	14.000	22 0	11,230	8,141	Three 46-in., three 86-in.	72	29,286	1,398	90	11,900	18.23
Aurania	Clydebank Works	1882	470	57 0	39 0	8.245	12.051	20 9	8,800	7,269	One 68-in., two 91 in.	72	23,284	1,001	90	8,500	17.5
Oregon	Fairfield Co.	1883	500	54 0	40 0	9.259	12.500	23 0	10,500	7,375	One 70-in., two 104-in.	72	38,047	1,428	110	13,300	18.3
America	Clydebank Works	1884	432	51 0	38 0	8.470	11 520	23 0	9,300	6,500	One 63-in., two 91 in.	66	22,750	882	95	7,354	17 8
Umbria	Fairfield Co.	1884	500	57 0	40 0	8 772	12.500	22 6	10,500	7,718	One 71-in., two 105-in.	72	38,817	1,606	110	14,321	20.18
Lahn	Fairfield Co.	1887	448	48 10	36 6	9.174	12.247	23 0	7,700	5,661	Two 32½ in., one 68-in., and two 85.in.	72	...	...	150	8.900	17.78
*Paris**	Clydebank Works	1888	528	63 0	41 10*	8 373	12.610	23 0	13,000	10,499	Two 45-in., two 71-in., and two 113.in.	60	50,265	1,293	150	20,100	21.8
Augusta Victoria	Vulcan Co., Stettin	1889	460	55 6	39 0	8.288	11.795	22 9	9,500	7,661	Two $41\frac{1}{16}$-in., two $66\frac{11}{16}$-in., and two $106\frac{5}{16}$-in.	63	36,000	1,120	150	14,110	18.31
Columbia	Laird Brothers	1889	463	55 6	39 0	8 333	11 860	22 9	9,500	7,578	Two 41-in., two 66-in., and two 101-in.	66	34,916	1,226	150	13,680	19.15
Teutonic	Harland & Wolff	1890	465	57 6	42 2	9.826	13.425	22 0	12,000	9,686	Two 43-in., two 68-in., and two 110-in	60	40,072	1,154	180	19,500	21
Normannia	Fairfield Co.	1890	500	57 3	38 0	8.73	13.150	22 0	10,500	8,716	Two 40-in., two 67-in., and two 106-in.	66	46,490	1,452	160	16,352	20.78
Spree	Vulcan Co., Stettin	1890	463	51 6	37 6	9.00	12.346	22 0	8,900	6,963	Two 38-in., one 75-in., and two 100-in.	72	...	...	165	13.000	19.6
Furst Bismarck	Vulcan Co., Stettin	1891	503	57 3	38 0	8 777	13 224	22 6	10,200	8,000	Two 43⅝-in., two 67-in., and two 106¼-in.	63	47,000	1,450	157	16,412	20 7
Campania	Fairfield Co.	1893	600	65 0	41 6	9.231	14 457	23 0	8,900	12,500	Four 37-in., two 79-in., and four 98-in.	60	82,000	2,630	165	30,000	22.01‡
St. Louis	Cramp, Philadelphia	1895	536	63 0	42 0	8.50	12.75	26 0	16,000	11,629	Four 28-in., two 55-in., two 77-in., four 77-in.	60	40,320	1,144	200	18,000	21.08‡
Kaiser Wilhelm der Grosse	Vulcan Co., Stettin	1897	625	66 0	43 0	9.46	14.27	28 0	20,880	14,349	Two 52-in., two 89-in., and four 96.4-in.	68.8	84,285	2,618	178	30,000	22.5 to 23‡
Oceanic	Harland & Wolff	1899	685	68 5	49 0	10.01	13.98	32 6	28,500	17,274	Two 47.5-in., two 79-in., and four 93-in.	72	74,686	1,962	192	27,000	20.72
Deutschland	Vulcan Co., Stettin	1900	662.9	67 0	44 0	9.89	15.06	29 0	23,620	16,502	Four 36.61-in., two 73.6-in., two 103.9-in., and four 106.3-in	72.8	85,468	2,188	220	36,000	23.25 to 23.5‡
Kronprinz Wilhelm	Vulcan Co., Stettin	1901	663†	66 0	43 0	...	..	29 0	21,300	15,000	Four 34.2-in., two 68.8-in., two 98.4-in., and four 102.3-in.	70.8	93,685	2,702	213	36,000	23 25 to 23.5‡
Kaiser Wilhelm II	Vulcan Co., Stettin	1903	678	72 0	52 6	9.41	12.9	29 0	26,000	20,000	Four 37.4-in., four 49.2-in., four 74.8-in., and four 112.2-in.	70.86	107,643	3,121	225	38,000	23 5‡
La Provence	St. Nazaire Works	1906	597 ft. 1½ in.	64 7½	41 8	9 23	14 35	26 9	19,160	13,750	Two 47.2-in., two 76.2.in., and four 88.18-in.	66.9	58,342	1,571	200	30,000	22.05
Lusitania	Clydebank Works	1907	769	88 0	60 4½	8.65	12.56	33 6	38,000	32,500	Turbines.	...	158,350	4,048	195	68,000	25

COAL CONSUMPTION OF ATLANTIC CUNARD LINERS.

......	*Britannia*, 1840.	*Persia*, 1856.	*Gallia*, 1879.	*Umbria*, 1884.	*Campania*, 1893.	*Lusitania*, 1907.
Coal necessary to steam to New York, tons.	570	1,400	836	1,900	2,900	5,000*
Cargo carried, tons.........	224	750	1,700	1,000	1,620	1,500
Passengers....................	115	250	320	1,225	1,700	2,198
Indicated horsepower.....	710	3,600	5,000	14,500	30,000	68,000
Steam pressure, pounds..	9	33	75	110	165	200
Coal per indicated horse-power per hour, pounds.	5.1	3.8	1.9	1.9	1.6	1.45*
Speed, knots...................	8.5	13.1	15½	19.0	22.0	25.0

* Estimated.

—"Engineering."

INERTIA AND TORSIONAL STRESSES AND PRESSURES ON BEARINGS, TOGETHER WITH AN INVESTIGATION OF THE LUBRICATION PROBLEM, OF THE PORT MAIN ENGINE U. S. S. *TENNESSEE.*

By Lewis Hobart Kenney, B. S., M. E. Associate.

The following computations and investigations were undertaken as a means of determining the causes of the heating of the crank-pin and main bearings, on the supposition that perhaps the pressures on the bearings were excessive. The crank pins have given the most trouble, while the main bearings have given but little. It is assumed that the engines were correctly lined up, and no allowance for friction of the engines has been made in the computations. The pressures on the crank pin are the resultant of the steam and inertia forces. One-half the connecting-rod weight has been considered concentrated at the crank pin, as a revolving weight, and its vertical component, corrected for weight due to gravity, has been added to the inertia force of the reciprocating weights for the crank-pin pressures. The revolving weight of the crank pin and crank webs has been considered in the computation of main bearing pressures.

The inertia forces of the valve gear were computed and the effect on the main bearings considered. This inertia force was reduced to an equivalent tangential force at crank radius, and a diagram was drawn to show the combined or resultant tangential force. The area included between this curve and its axis shows a positive area of 0.05 square inch. Of course, this should equal zero and the error may be largely graphical. This area reduced to a force at crank radius is

equal to 16.7 pounds which, compared to the mean tangential force of the engine, 268 000 pounds, makes the percentage error very small. All the computations were made by the writer, with the co-operation of Mr. William B. Robins.

The computations for bearing pressures give for the maximum crank-pin pressures 570 pounds per square inch of projected bearing surface, which is not excessive with efficient lubrication. Hence, after completing the computations the next subject to investigate was the system of lubrication.

The following analyses were made of the normal bearing metal used on the U. S. S. *Tennessee* and of "flakes" taken from the bearings. These flakes were found on the sides of the crank-pin brasses, near the distance pieces, and had apparently been drawn from the crown of the top and bottom brasses. Melting points and specific heats of the several elements are given, and, it seems, that the percentage of the elements of the flakes drawn from the normal metal bear a close relationship to their specific heats. The explanation suggested is that the bearing was not properly lubricated and, for a short time at maximum pressure, there was metallic contact. This condition, combined with the high surface velocity of the crank-pin surface (667 feet at 127 R.p.m.) generates heat and will raise the temperature of the bearing metal. The continued application of heat will melt the surface of the bearing metal and gradually draw it out into the form of flakes, which are deposited at the place of no bearing pressure—that is, near the distance pieces of the brasses.

ANALYSES OF ALLOYS *TENNESSEE* PORT H.P. CRANK PIN.

	Copper.	Tin.	Antimony.	Lead.
Specification	3.70	88.8	7.50	.0
Normal metal	3.70	89.57	6.66	0.07
Flake	1.12	94.80	3.95	0.14
Fused piece, $\frac{1}{64}$ inch thick	1.00	95.37	3.57	0.03
Melting point (F)	1929.	442.	810.	618.
Specific heat	0.0951	.0562	.0508	.0314

The oil film between the crank pin and its brass must sustain the full pressure of the piston load, otherwise we get metallic contact; and, if there is an opening in the brass at

this point of maximum film pressure, the film will relieve itself and flow out of the oil hole. This is illustrated by the oil flowing back in the oil tubes on the after face of the thrust shoes and flowing down the tube leading to the forward face of the thrust shoe. The oil films sustain the thrust in keeping the collar and shoe apart, and the oil hole provides a place for the film pressure to relieve itself; and the oil easily flows to the place of no bearing pressure. In the case of the crank pin, the oil is led to the top of the crank-pin brass and the point of film pressure is on the center line of the connecting rod; hence, from the above theory it would seem that this is not the best place for the oil holes. The centrifugal oiling device is feeding oil against the crank-pin oil film pressure during the upper part and the lower part of the stroke, which are points of maximum film pressure, and it would seem that this is not an especially efficient method. The pressure of the brasses alternates, which allows, to some extent, for admission of the lubricant. These oil tubes, as installed on the connecting rods, are open at the top for the drop feed, and the oil in these tubes would be subject to same inertia forces as the connecting rod itself. The inertia force is greater at the head end than at the crank end of the stroke and, this combined with the rapid alternations when running at high speed, would interfere with the gravity flow of the oil. There might be used a telescopic oiling device as fitted for the crosshead pin, to lead the oil to the crank pin. This would give a "head" from the top of the cylinder to the crank pin, which would overcome the inertia effect of the oil in the connecting-rod oil tubes and insure a positive flow to the crank pin, the oil for the crank pin to be led, instead of to the top of the brass as at present installed, to an oil reservoir formed by the distance pieces at the sides of the crank-pin brass, care being taken to fit the distance pieces so that they will be reasonably oil tight. This is the point of no bearing pressure; hence, no film pressure. The telescopic oil tube for this might be concentric with the crosshead-pin tube, and the casting on the crosshead pin would distribute the oil from

this center tube to the crank pin. This would make the device as compact as the present oiling device for the crosshead pin. The oil holes may be drilled diagonally through the upper brass from where the present oil pipes enter it, to these reservoirs. The holes in the top of the brass now used would be filled to agree with the theory that the oil film will relieve itself by flowing out of the holes. The oil will not flow against this film pressure unless it had a gravity "head," or was fed in under pressure in excess of the film pressure; this would apply to the down stroke, of course. The operation of this installation would be as follows: As the crank pin goes down, the oil in the inboard reservoir will be thrown against the crank pin by centrifugal force and adhesion to the crank pin will carry the oil over to the place of bearing pressure in the upper brass. On the up stroke, the oil in the other reservoir would be thrown against the crank pin by centrifugal force and adhesion will carry the oil around to lubricate the bottom brass, which is carrying the pressure.

In many cases an oil which saponifies is used. This saponified oil might accumulate in these distance-piece reservoirs and obstruct the supply of oil to the crank-pin. Of course, a mineral oil without any saponifying content might be used.

The several quantities computed are as follows:

Cylinders:

1. Resultant vertical component of the revolving weight.
2. Resultant vertical inertia force of reciprocating weight.
3. Resultant combined vertical force of 1 and 2.
4. No. 3 reduced to equivalent pressure per square inch of piston area.
5. No. 4 plus steam pressure reduced to total resultant effective pressure.
6. No. 5 reduced to connecting-rod force.
7. No. 6 reduced to pressure per square inch of crank-pin brass projected bearing surface.
8. No. 6 reduced to tangential force at crank radius.
9. No. 6 reduced to normal force at crank radius.
10. No. 5 reduced to crosshead pressure, total and per square inch of bearing surface.

Valve Gear:

1. Resultant vertical component of revolving weight.
2. Resultant vertical inertia force of reciprocating weight.
3. Resultant combined vertical force of 1 and 2.
4. No. 3 reduced to a tangential force at eccentric radius.
5. No. 4 reduced to equivalent force at crank radius.
6. No. 3 reduced to pressure per square inch of projected area of eccentric bearing surface.
7. For backing gear No. 1, 2, 3, 4, 5 and 6 were computed.

Main Bearings:

1. One-half the vertical effective force delivered by the piston.
2. Valve-gear resultant vertical force.
3. Vertical component of the centrifugal force of the crank pin and crank webs corrected for weight due to gravity.
4. Algebraic sum of 1, 2 and 3.
5. Algebraic sum of 1 for two cylinders if the bearing is between them.
6. 4 and 5 reduced to pressure per square inch of projected bearing surface.

The following symbols were used in the computations:

Cylinder.	Valve gear.	
A		Crank angle in degrees (0° at top of stroke).
	x	Eccentric angle in degrees (0° at top of stroke).
B		Connecting-rod angle in degrees.
	y	Eccentric-rod angle in degrees.
L		Length of connecting rod in feet $= 8.0$
	M	Length of eccentric rod in feet $= 8.25$
r		Radius of crank in feet $= 2.0$
	m	Radius of eccentric in feet $= 0.437$
n		Ratio, $\frac{L}{r} = 4$
	E	Ratio, $\frac{M}{m} = 18.86$

Cylinder.	Valve gear.	
F		Centrifugal force of one-half connecting-rod weight in pounds.
	v	Centrifugal force of eccentric and strap in pounds.
Fv		Vertical component of F.
	V	Vertical component of v.
G		Weight due to gravity of one-half connecting rod in pounds.
	l	Weight due to gravity of eccentric and strap in pounds.
F_C		Resultant vertical component of $F = G \pm F_V$
	V_V	Resultant vertical component of $v = l \pm V$
g		Weight due to gravity of reciprocating parts in pounds.
	H	Weight due to gravity of reciprocating parts in pounds.
f		Inertia force of reciprocating weights in pounds.
	K	Inertia force of reciprocating weights in pounds.
f_v		Vertical component of f.
	k	Vertical component of K.
P_R		Resultant vertical force of reciprocating weights, in pounds $= g \pm f_v$
	M	Resultant vertical force of reciprocating weights $= H \pm k$
P_F		Resultant vertical inertia force of all weights $= F_c + P_R$
	p	Resultant vertical inertia force of all weights $= M + V_v$
	t	Tangential force at eccentric radius equivalent to p.
	Q	Equivalent force of t at crank radius.
A_p		Area of top of piston in square inches.
a		Area of bottom of piston in square inches.
P_{FA}		P_F referred to piston area as equivalent pressure in pounds per square inch.
P_{Fa}		Effective pressure delivered by piston, in pounds per square inch.

Cylinder.	Valve gear.	
P		Effective pressure delivered by piston in pounds.
	E_p	Pressure per square inch of projected bearing area on eccentric due to p.
	h	Weight due to gravity of backing eccentric rod and one-half of link.
	q	Inertia force of backing gear.
	U	Vertical inertia force of backing gear.
	o	Resultant vertical force corrected for gravity of backing gear.
	O	Resultant vertical component of backing gear.
	t_e	Tangential force at eccentric radius equivalent to O.
	Q_e	Equivalent force of t_e at crank radius.
R		Pressure on connecting rod due to P.
T		Tangential pressure at crank pin due to R.
N		Normal pressure at crank pin due to R.
C		Crosshead pressure due to P.
C_a		Crosshead pressure per square inch of bearing surface.
K		Crank-pin brass pressure per square inch of projected area due to R.
P_d		Piston displacement from top of stroke in decimal of stroke.
W		A weight in pounds.
M		A mass corresponding to W.
D		Revolutions per minute.
s		Vertical component of centrifugal force of crank pin and crank web in pounds.
S		Vertical component of centrifugal force of crank pin and crank webs, in pounds, corrected for gravity.

COMPUTATIONS.

Derive formula for computing angle B for any angle A; refer to diagram on plot.

(1) $$\frac{L}{r} = n = 4 \text{ for this engine.}$$

$$DE = r \sin A = L \sin B$$

$$r \sin A = nr \sin B$$

(2) $$\sin B = \frac{\sin A}{n} = \frac{\sin A}{4}$$

Derive formulae for computing connecting rod, tangential, normal, and crosshead forces due to piston force P.

(3) $$R = \frac{P}{\cos B} = P \sec B$$

$$T = R \cos RDT$$

$$\text{angle } RDT = 180 - (A + B) - 90$$

$$= 90 - (A + B)$$

$$\cos (90 - (A + B)) = \sin (A + B)$$

(4) $$T = R \sin (A + B)$$

(5) $$= P\frac{\sin (A + B)}{\cos B}.$$

$$N = R \sin RDT$$

(6) $$= R \cos (A + B)$$

(7) $$= P \frac{\cos (A + B)}{\cos B}.$$

(8) $$C = P \tan B.$$

Derive piston displacement and inertia formulae.

Piston displacement is computed from the top of the stroke and for this position angles A and B are zero.

$$P_d = Ew + Ez$$

$$= r(1 - \cos A) + L(1 - \cos B)$$

(9) $$= r(1 + n - \cos A - n \cos B)$$

Subt. $n = 4$

(10) $$P_d = r(5 - (\cos A + 4 \cos B)).$$

In equation (9) substitute the value of $\cos B$ in terms of angle A.

From (2) $sin\ B = \dfrac{sin\ A}{n}$

$$sin\ B = (1 - cos^2 B)^{\frac{1}{2}}$$

(11) $$cos\ B = \frac{1}{n}(n^2 - sin^2 A)^{\frac{1}{2}}$$

Subst. Equa. (11) in 9

$$P_d = r\left(1 + n - cos\ A - (n^2 - sin^2 A)^{\frac{1}{2}}\right)$$

(Velocity) $$\frac{d\ P_d}{dA} = r\left(sin\ A + \frac{sinA\ cosA}{(n^2 - sin^2 A)^{\frac{1}{2}}}\right)$$

(12) (Acceleration)

$$\frac{d^2 P_d}{dA^2} = r\left(cos\ A + \frac{cos^2 A - sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}} + \frac{sin^2 A\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right)$$

(13) $$= r\left(cos\ A + \frac{n^2\ cos^2\ A}{(n^2 - sin^2 A)^{\frac{3}{2}}} - \frac{sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}}\right)$$

(14) $$\text{Let } Z_1 = \frac{n^2\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}} - \frac{sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}}$$

(15) $$\text{then, } \frac{d^2\ P_d}{d\ A^2} = r\,(cos\ A + Z_1).$$

Equation (12) will reduce to

$$\frac{d^2\ P_d}{d\ A^2} = r\left(cos\ A + \frac{n^2\ cos^2 A - sin^2 A\ (n^2 - sin^2 A)}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right)$$

(16) $$\text{Let } Z_2 = \frac{n^2\ cos^2 A - sin^2 A\ (n^2 - sin^2 A)}{(n^2 - sin^2 A)^{\frac{3}{2}}}.$$

The value Z_2 may be a little more convenient for computation than Z_1.

Equation (12) will also reduce to

$$\frac{d^2\ P_d}{d\ A^2} = r\left(cos\ A + \frac{cos\ 2\ A}{(n^2 - sin^2 A)^{\frac{1}{2}}} + \frac{sin^2 A\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right).$$

If n is very large, as in the case of eccentric rods, $sin\ A$ will never be very large so that the last term may be omitted, and the second term will reduce to $\dfrac{cos\ 2A}{n}$. The greater the

value of n the nearer this approximate value will be to the value of Z_1 or Z_2 and (12) reduces to

$$(17) \qquad \frac{d^2 P_d}{d A^2} = r\left(\cos A + \frac{\cos 2A}{n}\right).$$

The angular velocity $= \frac{dA}{dt} = \frac{2 \pi D}{60}$

$$(18) \qquad \frac{dA^2}{dt^2} = \left(\frac{\pi D}{30}\right)^2$$

Subst. (18) in (15).

$$(19) \qquad \frac{d^2 P_d}{d t^2} = \left(\frac{\pi D^2}{30}\right) r(\cos A + Z_1).$$

We have the general formula: force = mass × acceleration.

Substituting (19) in this formula

$$\text{Force} = M\left(\frac{\pi D}{30}\right)^2 r(\cos A + Z_1)$$

Substituting $\frac{W}{g} = M$;

$$\text{Force} = \frac{\pi^2}{900 g} W r D^2 (\cos A + Z_1)$$

$$(20) \qquad = 0.000\,341\ W r D^2 (\cos A + Z_1)$$

The total weight for the reciprocating force is the sum of the weights—piston, piston rod, crosshead, and one-half the connecting rod. The other half of the connecting rod is considered concentrated at the crank pin as a revolving weight.

The inertia of the reciprocating weights is computed by (20) and corrected for the weight due to gravity; the centrifugal force of the revolving weight is computed and its vertical component is corrected for the weight due to gravity. The algebraic sum of these two quantities is the total resultant inertia force.

The computations are all mathematical and only the results graphical, in order to insure accuracy and speed. By

dividing the connecting-rod weight into one-half as a reciprocating weight and the other as a revolving weight concentrated at the crank pin makes a convenient method of disposing of this troublesome problem. This method is used by Naval Constructor D. W. Taylor, U. S. N., in Volume 9, Transactions Society of Naval Architects and Marine Engineers; Sanford A. Moss' paper, "Influence of Connecting Rod upon Engine Forces," Volume 26, Transactions American Society of Mechanical Engineers, and Dr. D. S. Jacobus' paper, "Counter Weights for Large Engines," Volume 26 of the same society.

The mathematics for this investigation are developed from Dr. D. S. Jacobus' paper which appeared in Volume 11, Transactions of the American Society of Mechanical Engineers.

CYLINDER WEIGHTS, IN POUNDS.

Reciprocating.	H.P.	I.P.	L.P.
Piston	1 852	3 485	4 300
Piston rod	988	988	833
Crosshead	3 112	3 112	2 246
½ Connecting rod	3 600	3 600	2 940
Total	9 552	11 185	10 319
For computations (Total)	9 550	11 190	10 320
Revolving weight	3 610	3 610	2 940

VALVE GEAR.

The weight of the valve gear was divided into a reciprocating and revolving weight. The revolving weight was the eccentric and eccentric straps; the eccentric rod, one-half the suspension link, and all the reciprocating parts were taken as reciprocating weights for the ahead valve gear. For the backing gear the revolving weight was the same as above and the eccentric rod and other half the suspension link as the reciprocating weight. There may be an error in this assumption but perhaps it is sufficiently approximate for the purpose.

WEIGHT, IN POUNDS OF VALVE GEAR.

	Total.	AHEAD. Recip-rocat-ing.	AHEAD. Revolv-ing.	ASTERN. Recip-rocat-ing.	ASTERN. Revolv-ing.
H.P. CYLINDER					
1 Eccentric	700	...	700	...	700
1 Eccentric strap	740	...	740	...	740
1 Eccentric rod	375	375	...	375	...
1 Piston valve	1 155	1 155	...	...	...
1 Valve stem and balance piston	220	220	...	...	...
1 Valve-stem crosshead	200	200	...	...	...
1 Link	550	275	...	275	...
Total	3 940	2 225	1 440	650	1 440
Upward force of balance piston	...	1 902	...	...	...
Unbalanced force	...	323	...	...	...
I.P. CYLINDER	5 480	3 765	1 440	650	1 440
Upward force of balance piston	...	3 804	...	...	...
Excess unbalanced force	...	39	...	...	...
L.P. CYLINDER	5 920	4 205	1 440	650	1 440
Upward force of balance piston	...	5 474	...	...	...
Excess unbalanced force	...	1 269	...	...	...

CURVES PLOTTED.

The first curves drawn are for the piston displacement. The H.P. is drawn from the top of its stroke through one revolution, and the other cylinders are laid down in proper crank phase. The ratios of the displacements are equal to the cylinder volume ratios; the clearances and receiver volumes are ratios of H.P. cylinder volume. Valve displacement curves laid down in proper phase, together with the position of the port, show the position of cut off, release, compression and admission. The corresponding piston displacement may be scaled from the piston displacement curve. The volume the steam occupies may be scaled from the piston displacement curves; adding the clearance and receiver volumes will give the total volume of steam.

The valve displacement curve is plotted from equation (9) computed from top of stroke when x and y = zero.

$$P_d = r(1 + n - \cos A - n \cos B)$$

Substituting in (9),

Valve displacement $= m\,[19.86 - (\cos x \quad 18.86 \cos y)]$

This curve is compared to a sine curve as also is the piston

displacement, in the computed data, to determine the error if a sine curve were used.

After the piston and valve displacement curves had been laid down the actual indicator cards were plotted to correspond to the crank angles. Beginning at the top of the stroke the steam line of the top card is drawn for 180° and from there on the back pressure line of the card, in the same way the bottom card is drawn. The difference between these two lines is the effective pressure.

REDUCTION OF INDICATOR CARD PRESSURE TO EQUIVALENT PRESSURE PER SQUARE INCH OF PISTON.

Symbols:

Top P = Absolute pressure from indicator card.

A = Area of piston in square inches.

P_1 = Equivalent effective pressure per square inch of piston.

Bottom p = Absolute pressure from indicator card corresponding to P.

a = Area of piston in square inches.

p_1 = Equivalent effective pressure per square inch of piston.

K = Effective load on piston.

r = Ratio $\dfrac{A}{a}$

$$PA - pa = K$$

$$(Pr - p)a = K$$

$$P_1 = \frac{K}{A}; \qquad p_1 = \frac{K}{a}$$

$$= \frac{(Pr - p)a}{A} \qquad = \frac{(Pr - p)a}{a}$$

$$= P - \frac{p}{r} \qquad = Pr - p$$

H.P.	$P_1 = P - .950p$	$p_1 = 1.052P - p$
I.P.	$= P - .985p$	$= 1.017P - p$
L.P.	$= P - .987p$	$= 1.013P - p$

This method will give the correct effective pressure, but the curves as drawn give only the actual difference between the steam pressure of one card and the corresponding back pressure of the other. The result is that the top pressures are a little less and the bottom pressures are a little more, than the correct effective pressure. The mean of the H.P. error for the top and bottom is 5.5 pounds and the error of the other cylinders will of course be less.

REDUCTION OF INERTIA PRESSURE TO AN EQUIVALENT PRESSURE PER SQUARE INCH OF PISTON.

The total inertia force was first computed and the force exerted upward was divided by the area of the top of the piston. This will give the pressure required to cushion the inertia force and the pressure required to accelerate the piston on its return stroke. The pressure on the bottom of the piston was computed by the same method. The inertia curve laid down from this data will be fair, with no change at ends of stroke, and it is necessary to compute values for only 180° instead of 360°. The method involves a slight error because, starting from the top of the stroke, the inertia force is negative and about midstroke changes to a positive force, hence the inertia force on the down stroke should be divided by the top area and the bottom pressure determined by the same method. This pressure added (algebraically) to the indicator-card pressure will give the total effective pressure acting on the top or bottom of the piston. The inertia curves are correct from the beginning of the stroke to the point where the inertia force changes sign, and from there to the end of the stroke are in error to the extent of the ratio of the top and bottom areas of the piston. The error in determining the effective steam pressure by the method used is partly compensated for by the error in the inertia curve. But the error is not very great and becomes less as the piston areas increase.

After determining the effective piston force, the pressures on the crossheads and crank pin were computed. The maximum pressure in pounds per square inch of crank-pin projected area are as follows:

	Top.	Bottom.
H.P.	560	500
I.P.	570	460
F.L.P.	400	410
A.L.P.	400	400

Mean velocity, in feet per minute, of rubbing surface in bearing:

Crank pin	667
Main bearing	616

By the above it will be seen that the crank-pin brasses are subjected to very heavy pressures twice per second—127 R.p.m. is the condition selected for this computation—and the viscosity of some lubricants may not be sufficient to maintain a film between the crank pin and its brasses. If the film breaks down there will result for an instant, perhaps, heating and dragging of the bearing metal. It is also evident from the diagram that no change in the valve settings would cushion the inertia forces, because the inertia is greater than the available steam pressure; in the case of the L.P. cylinders, the crank has revolved an angle of about 50 or 60 degrees before the steam pressure balances the inertia.

The inertia curve for 100 R.p.m. has been plotted because the graphical interpretation is so much easier than the statement that the inertia varies as the square of the revolutions.

MAIN BEARINGS.

The computation for the main bearing pressures is by the following arbitrary method: The shaft is assumed to fit closely in its bearings and slightly elastic so that a light stress will put the shaft against its bearings. Hence, one-half of the effective piston force plus one-half of the resultant, vertical component of the crank-pin and crank-web centrifugal force is assumed to be distributed on each bearing. The resultant force of the valve gear is added algebraically to its cylinder crank-bearing pressure. In the case of a bearing between two cranks, the algebraic sum of one-half the effective piston forces plus the crank-pin and crank-web forces, as above, is taken. The pressure is reduced to the pressure per square inch of projected area of bearing surface. The curves

drawn show the vertical force for the crank angle and not the normal pressure at crank radius. This method does not, of course, take into consideration the torsional effect on the crank shaft or the moments set up by the valve and piston pressures.

For convenience in computing, the following factors were determined and computation forms made:

1 B

2 $\frac{1}{\cos B}$

3 $\sin (A + B)$

4 $\cos (A + B)$

5 $\tan B$

6 $5 - (\cos A + 4 \cos B)$

7 $\cos A$

8 $\cos A + Z_1$ was taken from Foley's Mechanical Engineers' Reference Book, and the values check very closely with those given by Naval Constructor D. W. Taylor, U. S. N., in Volume 9 of the Transactions Society of Naval Architects and Marine Engineers.

9 y

10 $\cos x$

11 $\cos x + \frac{\cos 2x}{E}$ Equation 17 with substitutions.

12 $\frac{\sin (x + y)}{\cos y}$ Equation 5 with substitutions.

13 $19.86 - (\cos x + 18.86 \cos y)$ Equation 9 with substitutions.

H.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Cos A	1.000 00	.984 81	.906 31	.707 11	.422 62	.173 65	.000 00	—.173 65	—.422 62	—.707 11	—.906 31	—.984 81	—1.000 00
$F_v = F \cos A$	39 650	39 000	35 900	28 030	16 750	6 890	0	—6 890	—16 750	—28 030	—35 900	—39 000	—39 650
$F_e = G \pm F_v$	36 040	35 390	32 290	24 420	13 140	3 280	—3 610	—10 500	—20 360	—31 640	—39 510	—42 610	—43 260
Multiplier	1.250 0	1.220 9	1.070 5	.711 4	.260 3	—.068 3	—.258 4	—.415 6	—.584 7	—.702 9	—.742 6	—.749 2	—.750 0
$f_v = f \times$ mult.	131 200	128 200	112 400	74 700	27 330	—7 170	—27 110	—43 650	—61 400	—73 800	—78 000	—78 600	—78 750
$P_R = g \pm f_v$	121 650	118 650	102 850	65 150	17 780	—16 720	—36 660	—53 200	—70 950	—83 350	—87 550	—88 150	—88 300
$P_F = P_R + F_e$	157 690	154 040	135 140	89 570	30 920	—13 440	—40 270	—63 700	—91 310	—114 990	—127 060	—130 760	—131 560
P_{FA}	135.6	132 4	116.1	77.0	26.6	—12.1	—36.3	—57.6	—82.5	—103.8	—114.8	—118.3	—118.9
P_{Fa} down	—50	—6	26	62	101	135	157	176	195	200	180	153	118
P_{Fa} up	111	158	185	185	153	116	96.0	79.0	65.0	53.0	42.0	29	0
P down.	—55 200	—6 650	30 250	72 200	117 500	157 100	182 700	204 800	227 000	232 800	209 400	178 200	137 400
P up	122 800	175 000	210 000	210 000	169 500	128 500	106 300	87 500	72 000	58 700	46 500	32 100	0
$\frac{1}{\cos B}$	1.000 00	1.000 94	1.005 63	1.016 00	1.026 70	1.031 76	1.032 80	1.031 76	1.026 70	1.016 00	1.005 63	1.000 94	1.000 00
$R = \frac{P}{\cos B}$ down	—55 200	—6 650	30 450	73 400	120 700	162 200	188 700	211 200	233 000	236 500	211 000	178 200	137 400
$R = \frac{P}{\cos B}$ up	122 800	175 000	211 000	213 400	174 000	132 600	109 600	90 200	73 800	59 600	46 800	32 100	0
Sin (A + B)	0.000 00	.216 24	.516 01	.820 97	.978 50	.997 25	.968 24	.911 74	.786 98	.570 97	.324 50	.130 73	.000 00
T = R sin (A + B) down	0	—1 440	15 700	60 300	118 100	161 600	182 600	192 600	183 500	135 000	68 500	23 290	0
T = R sin (A + B) up	0	37 850	108 900	175 200	170 200	132 100	106 200	82 300	58 100	34 050	15 170	4 195	0
$K = \frac{R}{430}$ down	—128.4	—15.5	70.8	170.8	280.6	377.0	438.7	491.0	542.0	550.0	490.6	414.5	319.5
$K = \frac{R}{430}$ up	285.3	406.7	490.5	496.0	404 5	308.2	254 9	210.0	171.6	138.5	108.8	74.6	0
Tan B	0.000 00	.043 45	.106 25	.179 61	.232 63	.254 02	.258 20	.254 02	.232 63	.179 61	.106 25	.043 45	000 00
C = P tan B down	0	—289†	3 215	12 960	27 340	39 880	47 200	52 000	52 700	41 800	22 210	7 740	0
C = P tan B up	0	7 560	22 320	37 700	39 400	32 610	27 420	22 210	16 740	10 550	4 950	1 395	0
Ca — down	0	—0 47†	3.5	14.1	29 8	43.4	51.4	56.6	57.4	45.5	24 2	8.4	0
Ca — up	0	8.28	24.3	41.0	41.9	35 5	29.8	24.2	18.2	11.5	5.4	1.5	0
Cos (A + B)	1.000 00	.976 34	.856 59	.570 97	.206 27	.074 16	—.250 00	—.410 76	—.616 98	—.820 97	—.945 89	—.991 42	—1.000 00
N = R cos (A + B) down	—55 200	—6 490	26 100	41 900	24 900	12 020	—47 150	—86 900	—143 700	—194 200	—199 500	—176 500	—137 400
N = R cos (A + B) up	—122 800	—170 700	—180 800	—122 000	—35 900	—9 840	27 400	37 080	45 500	48 950	44 250	31 820	0
Piston displacement from top = 5 — (4 cos B + cos A)	0 000 00	.018 95	.116 10	.355 89	.681 42	.949 47	1.127 00	1.296 77	1.526 66	1.770 11	1.928 71	1.988 57	2.000 00
Sine curve	0.000 00	.015 19	.093 69	.292 89	.577 38	.826 35	1.000 00	1.173 65	1.422 62	1.707 11	1.906 31	1.984 81	2.000 00
Difference	0	.003 76	.022 41	.063 00	.104 04	.123 12	.127 00	.123 12	.104 04	.063 00	.022 40	.003 76	0
Angle B	0° 0′ 0″	2° 29′ 16.8″	6° 3′ 53.0″	10° 10′ 56.3″	13° 5′ 45.0″	14° 15′ 9.9″	14° 28′ 39.1″	14° 15′ 9.9″	13° 5′ 45.0″	10° 10′ 56.3″	6° 3′ 53.0″	2° 29′ 16 8″	0° 0′ 0″

† On backing face.

I.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_v = F \cos A$													
*$F_e = G \pm F_v$													
*Multiplier													
$f_v = f \times$ mult.	153 800	150 200	131 800	87 600	32 050	−8 410	−31 800	−51 200	−72 000	−86 500	−91 400	−92 100	−92 300
$P_R = g \pm f_v$	142 610	139 010	120 610	76 410	20 860	−19 600	−42 990	−62 390	−83 190	−97 690	−102 590	−103 290	−103 490
$P_T = P_R + F_e$	178 650	174 400	152 900	100 830	34 000	−16 320	−46 600	−72 890	−103 550	−129 330	−142 100	−145 900	−146 750
P_{TA}	56.4	55.0	48.3	31.8	10.7	−5.2	−15.0	−23.5	−33.3	−41.6	−45.7	−46.9	−47.2
P_m { down	−14.0	−3.0	6.5	22.0	34.0	45.5	55.0	63.0	72.0	75.0	64.0	51.5	39.0
P_m { up	39.0	50.0	58.0	63.0	58.5	46.0	37.0	28.0	18.0	13.5	10.0	3.0	−13.0
P { down	−43 550	−9 330	20 580	69 700	107 600	144 000	174 200	190 400	228 000	237 300	202 500	163 000	123 500
P { up	121 300	155 500	180 400	196 000	182 000	143 200	115 200	87 100	56 000	42 000	31 100	9 330	−41 150
* $\frac{1}{\cos B}$													
$R = \frac{P}{\cos B}$ { down	−43 550	−9 340	20 690	70 850	110 400	148 500	180 000	205 800	234 100	241 100	203 500	163 200	123 500
$R = \frac{P}{\cos B}$ { up	121 300	155 600	181 500	199 200	187 000	147 600	119 000	89 900	57 600	42 650	31 300	9 340	−41 150
*Sin (A + B)													
$T = R \sin(A+B)$ { down	0	−2 020	10 670	58 200	108 000	148 000	174 300	187 600	184 200	137 600	66 100	21 300	0
$T = R \sin(A+B)$ { up	0	33 650	93 700	163 500	183 000	147 200	115 200	82 000	45 300	24 350	10 150	1 220	0
$K = \frac{R}{430}$ { down	−101.0	−21.7	48.0	165.0	257.0	345.0	419.0	479.0	544.0	561.0	473.0	379.0	287.0
$K = \frac{R}{430}$ { up	282.0	362.0	422.0	463.0	435.0	343.0	277.0	209.0	134.0	99.0	73.0	21.7	−96.0
*Tan B													
$C = P \tan B$ { down	0	−405†	2 190	12 500	25 050	36 360	44 950	50 650	53 100	42 650	21 580	7 080	0
$C = P \tan B$ { up	0	6 755	19 160	35 200	42 320	36 350	29 750	22 100	13 020	7 550	3 310	405	0
Ca — { down	0	−0.6†	2.4	13.6	27.3	39.8	49.0	55.2	57.8	46.5	23.4	7.7	0
Ca — { up	0	7.4	20.9	38.3	46.1	39.6	32.4	24.1	14.2	8.2	3.6	0.4	0
*Cos (A + B)													
$N = R \cos(A+B)$ { down	−43 550	−9 120	17 720	40 450	22 780	11 010	−45 000	−84 550	−144 500	−198 000	−192 400	−161 600	−123 500
$N = R \cos(A+B)$ { up	−121 300	−151 800	−155 400	−113 700	−38 600	−10 940	29 750	36 920	35 550	35 040	29 600	9 260	−41 150
*Piston displacement from top $= 5 - (4 \cos B + \cos A)$													
*Sine curve													
*Difference													
*Angle B													

* The values are the same for each cylinder.

† On backing face.

F.L.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_v = F \cos A$	32 400	31 900	29 360	22 900	12 700	5 630	0	—5 630	—13 700	—22 900	—29 360	—31 900	—32 400
*$F_e = G \pm F_v$	29 460	28 960	26 420	19 960	9 760	2 690	—2 940	—8 570	—16 640	—25 840	—32 300	—34 840	—35 340
*Multiplier													
$f_v = f \times$ mult.	141 800	138 600	121 600	80 800	29 550	—7 750	—29 320	—47 200	—66 400	—79 800	—84 400	—85 000	—85 200
$P_R = g \pm f_v$	131 480	128 280	111 280	70 480	19 230	—18 070	—39 640	—57 520	—76 720	—90 120	—94 720	—95 320	—95 520
$P_F = P_R + F_e$	160 940	157 240	137 700	90 440	28 990	—15 380	—42 580	—66 090	—93 360	—115 960	—127 020	—130 160	—130 860
P_{FA}	37.3	36.4	31 9	21.0	6.7	—3 6	—10.0	—15.5	—21.9	—27.2	—29.8	—30 5	—30.7
P_m down	—33.5	—27.0	—19.0	—5.4	10 0	21.0	27.5	33.3	38.5	39.4	37.7	34 4	30.7
P_m up	36.5	41.7	41.5	37.5	28.0	18.3	12.5	7.8	0	—5.0	—11.0	—15.8	—21.5
P down.	—144 500	—116 500	—82 000	—23 300	43 150	90 600	118 600	143 600	166 200	170 100	162 700	148 400	132 500
P up	155 400	177 500	176 600	159 700	119 200	77 900	53 200	33 200	0	—21 300	—46 850	—67 300	—91 500
* $\frac{1}{\cos B}$													
$R = \frac{P}{\cos B}$ down	—144 500	—116 600	—82 500	—23 690	44 300	93 500	122 500	148 200	170 500	172 800	163 600	148 500	132 500
$R = \frac{P}{\cos B}$ up	155 400	177 600	177 600	162 200	122 400	80 300	54 900	34 250	0	—21 650	—47 100	—67 400	—91 500
*Sin (A + B)													
$T = R \sin(A+B)$ down	0	—25 220	—42 570	—19 440	43 350	93 200	118 500	135 200	134 200	98 700	53 200	19 410	0
$T = R \sin(A+B)$ up	0	38 400	91 600	133 200	119 800	80 000	53 200	31 230	0	—12 360	—15 310	—8 810	0
$K = \frac{R}{430}$ down	—336.0	—271.0	—192.0	—55.1	103.1	217.5	285.0	344 2	396.2	402.0	380.5	315.5	308 0
$K = \frac{R}{430}$ up	361.2	413.0	413.0	377.0	284.5	186.8	127.8	79.7	0	—50.3	—109.6	—156.8	—213.0
*Tan B													
$C = P \tan B$ down	0	—5 060†	—8 720†	—4 185†	10 040	23 010	30 620	36 450	38 640	30 520	17 280	6 450	0
$C = P \tan B$ up	0	7 710	18 760	28 650	27 720	19 770	13 730	8 430	0	—3 825†	—4 985†	—2 925†	0
Ca — down	0	—17.4†	—19.6†	—9.4†	16.4	37.6	50.0	59.5	63.1	49.9	28.2	10 5	0
Ca — up	0	12.6	30.6	46.8	45.3	32.3	22.4	13.8	0	—8.6†	—11 2†	—6.6†	0
*Cos (A + B).													
$N = R \cos(A+B)$ down	—144 500	—113 800	—70 700	—13 530	9 150	6 940	3 060	60 900	105 100	141 800	154 700	147 100	132 500
$N = R \cos(A+B)$ up	155 400	173 300	152 100	92 600	25 230	5 960	13 720	14 070	0	—17 780	—44 520	—66 750	—91 500
*Piston displacement from top $= 5 - (4 \cos B + \cos A)$													
*Sine curve													
*Difference													
*Angle B													

* The values are the same for each cylinder.

† On backing face.

A.L.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_v = F \cos A$													
*$F_e = G \pm F_v$													
*Multiplier													
*$f_v = f \times$ mult.													
*$P_R = g \pm f_v$													
*$F_P = P_R + F_e$													
*P_{FA}													
P_{Fa} down	−30.0	−25.0	−16.9	−3 0	11.9	22 0	27.9	32.8	37.8	39 4	35 8	33.3	28 5
P_{Fa} up	37.5	40 5	40 0	35.5	26.5	18.5	13.3	8.0	1.5	−4.5	−9.4	−13.3	−18.0
P down	−129 500	−107 900	−72 900	−12 950	51 400	95 000	120 400	141 500	163 200	170 100	158 800	143 600	123 000
P up	159 700	172 500	170 300	151 100	112 800	78 800	56 600	34 070	6 380	−19 160	−40 020	−56 600	−76 600
*$\frac{1}{\cos B}$													
$R = \frac{P}{\cos B}$ down	−129 500	−108 000	−73 300	−13 160	52 800	98 000	124 400	146 000	167 500	172 800	159 600	143 700	123 000
$R = \frac{P}{\cos B}$ up	159 700	172 600	171 300	153 500	115 800	81 300	58 500	35 150	6 550	−19 460	−40 270	−56 700	−16 600
*Sin (A + B)													
$T = R \sin(A+B)$ down	0	−23 350	−37 810	−10 810	51 700	97 700	120 400	133 200	131 800	98 700	51 900	18 770	0
$T = R \sin(A+B)$ up	0	37 300	88 400	126 000	113 400	81 000	56 600	32 050	5 150	−11 110	−13 080	−7 410	0
$K = \frac{R}{430}$ down	−301.0	−251 0	−170.6	−30 6	122 8	228.0	289.0	339.0	389.0	402 0	371.0	334 0	286 0
$K = \frac{R}{430}$ up	371.0	401.0	398.0	357.0	269.0	189.2	136.0	81.7	15.2	−45.3	−93 6	−132.0	−178.3
*Tan B													
$C = P \tan B$ down	0	−4 690†	−7 750†	−2 325†	11 950	24 120	31 100	35 920	37 940	30 520	16 880	6 240	0
$C = P \tan B$ up	0	7 500	18 110	27 120	26 240	20 200	14 620	8 655	1 484	−3 441†	−4 255†	−2 460†	0
Ca — down	0	−10.6†	−17.5†	−5 2†	19.5	39.4	50 8	58.7	62.0	49 9	27 6	10 2	0
Ca — up	0	12.3	29.6	44.3	42.9	33.0	23.9	14.2	2 4	−7.7†	−9.6†	−5.5†	0
*Cos (A + B)													
$N = R \cos(A+B)$ down	−129 500	−105 300	−62 800	7 520	10 900	7 270	−31 100	−60 000	−103 300	−141 800	−150 900	−142 300	−123 000
$N = R \cos(A+B)$ up	−159 700	−168 400	−146 700	−87 600	−23 900	−6 030	14 620	14 440	4 040	15 980	38 080	56 150	76 600
*Piston displacement from top $= 5 - (4 \cos B + \cos A)$													
*Sine curve													
*Difference													
*Angle B													

* The values are the same for each cylinder.

† On backing face.

H.P. VALVE GEAR.

Angle x	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Cos x	1.000 00	.984 81	.906 31	.707 11	.422 62	.173 65	.000 00	—.173 65	—.422 62	—.707 11	—.906 31	—.984 81	—1.000 00
$V = v \cos x$	3 470	3 418	3 143	2 453	1 466	603	0	—603	—1 466	—2 453	—3 143	—3 418	—3 470
$V_v = l \pm V$	2 030	1 978	1 703	1 013	26	—837	—1 440	—2 043	—2 906	—3 893	—4 583	—4 858	—4 910
Multiplier	1.053 02	1.034 63	.940 39	.707 11	.388 54	.123 83	—.053 02	—.223 47	—.456 70	—.707 11	—.872 23	—.934 99	—.946 98
$k = K \times$ mult.	5 640	5 540	5 035	3 785	2 080	663	—284	—1 197	—2 445	—3 785	—4 670	—5 010	—5 020
$M = H \pm k$	5 317	5 217	4 712	3 462	1 757	340	—607	—1 520	—2 768	—4 108	—4 993	—5 333	—5 343
$p = M + V_v$	7 347	7 195	6 415	4 475	1 783	—497	—2 047	—3 563	—5 674	—8 001	—9 576	—10 191	—10 253
$\frac{\sin(x+y)}{\cos y}$	0.000 00	.182 71	.442 94	.733 64	.926 64	.993 88	1.000 00	.975 73	.885 98	.680 58	.402 30	.164 58	.000 00
$t = \frac{p \sin(x+y)}{\cos y}$	0	1 314	2 840	3 288	1 652	—494	—2 047	—3 480	—5 030	—5 450	—3 850	—1 678	0
$Q = p \frac{m}{r}$	0	288	622	719	362	—108	—448	—762	—1 100	—1 192	—842	—367	0
$E_p = \frac{p}{178}$	41.3	40.5	36.0	25.2	10.0	—2.8	—11.5	—20.0	—31.9	—45.0	—53.8	—57.2	—57.6
Valve displacement from top = 19 86 —(cos x+18.86 cos y)	0.00	.01	.10	.30	.60	.85	1.03	1.19	1.44	1.72	1.92	1.99	2.00
Sine curve	0.00	.015	.094	.293	.577	.826	1.00	1 174	1.423	1.707	1.906	1 985	2.00
Scale for plotting	0.00	.01	.05	.16	.32	.45	.54	.64	.76	.90	1.01	1.05	1.05
Angle y	0° 0′ 0″	0° 31′ 39.3″	1° 17′ 2 4″	2° 8′ 55.2″	2° 45′ 15.8″	2° 59′ 35.6″	3° 2′ 21.8″	2° 59′ 35.6″	2° 45′ 15 8″	2° 8′ 55.2″	1° 17′ 2.4″	0° 31′ 39.3″	0° 0′ 0″

I.P. VALVE GEAR.

Angle x	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos x													
*$V = v \cos x$													
*$V_v = l \pm V$													
*Multiplier													
$k = K \times$ mult.	9 540	9 370	8 520	6 410	3 522	1 123	—481	—2 050	—4 140	—6 410	—7 910	—8 480	—8 580
$M = H \pm k$	9 580	9 410	8 560	6 450	3 562	1 163	—441	—2 010	—4 100	—6 370	—7 870	—8 440	—8 540
$p = M + V_v$	11 610	11 388	10 263	7 463	3 588	—326	—1 881	—4 053	—7 006	—10 263	—12 453	—13 298	—13 450
*$\frac{\sin (x + y)}{\cos y}$													
$t = \frac{p \sin (x + y)}{\cos y}$	0	2 080	4 540	5 480	3 325	—324	—1 881	—3 955	—6 200	—6 990	—5 010	—2 190	0
$Q = p \frac{m}{r}$	0	455	994	1 198	728	—71	—412	—866	—1 356	—1 528	—1 096	—479	0
$E_p = \frac{p}{178}$	65.2	64.0	57.6	41.9	20.2	—1.9	—10.6	—22.8	—39.3	—57.6	—70.0	—74.7	—75.6
*Valve displacement from top = 19.86 — (cos x + 18.86 cos y)													
*Sine curve													
*Scale for plotting													
*Angle y													

*The values are the same for each cylinder.

F.L.P. AND A.L.P. VALVE GEAR.

Angle x	0 / 360	10 / 350	25 / 335	45 / 315	65 / 295	80 / 280	90 / 270	100 / 260	115 / 245	135 / 225	155 / 205	170 / 190	180
*Cos x													
*$V = v \cos x$													
*$V_r = l + V$													
*Multiplier													
$k = Kx$ mult.	10 670	10 480	9 520	7 160	3 940	1 255	—537	—2 264	—4 625	—7 160	—8 840	—9 480	—9 600
$M = H + k$	11 940	11 750	10 790	8 430	5 270	2 525	633	—994	—3 355	—5 890	—7 570	—8 210	—8 330
$p = M + V_r$	13 970	13 728	12 493	9 443	5 236	1 688	—807	—3 037	—6 261	—9 783	—12 153	—13 068	—13 240
*$\frac{\sin(x+y)}{\cos y}$													
$t = \frac{p \sin(x+y)}{\cos y}$	0	2 510	5 520	6 930	4 860	1 678	—807	—2 963	—5 550	—6 660	—4 890	—2 150	0
$Q = p \frac{m}{r}$	0	550	1 207	1 516	1 063	367	—177	—649	—1 215	—1 456	—1 070	—471	0
$E_p = \frac{p}{178}$	78.5	77.1	70.1	53.1	29.4	9.5	4.5	17.1	35.2	54.9	68.3	73.4	74.3
*Valve displacement from top = 19.86 — (cos x + 18.86 cos y)													
*Sine curve													
*Scale for plotting													
*Angle y													

* The values are the same for each cylinder.

H.P., I.P., F. AND A.L.P.——BACKING VALVE GEAR.

	0 / 360	10 / 350	25 / 335	45 / 315	65 / 295	80 / 280	90 / 270	100 / 260	115 / 245	135 / 225	155 / 205	170 / 190	180
$U = q x$ mult.	1 648	1 616	1 470	1 105	608	194	—83	—350	—714	—1 105	—1 364	—1 462	—1 481
$o = h \pm U$	998	966	820	455	—42	—456	—733	—1 000	—1 364	—1 755	—2 014	—2 112	—2 131
$O = o + V_r$	3 028	2 944	2 523	1 468	—16	—1 293	—2 173	—3 043	—4 270	—5 648	—6 597	—6 970	—7 041
$t_o = \frac{o \sin(x+y)}{\cos y}$	0	538	1 118	1 077	—15	—1 285	—2 173	—2 966	—3 782	—3 850	—2 652	—1 147	0
$Q_o = O \frac{m}{r}$	0	118	244	236	—3	—281	—476	—649	—828	—842	—580	—251	0

MAIN BEARING PRESSURES.

Downward forces are +
Upward forces are —

Pressures in pounds.
Projected bearing area is used.

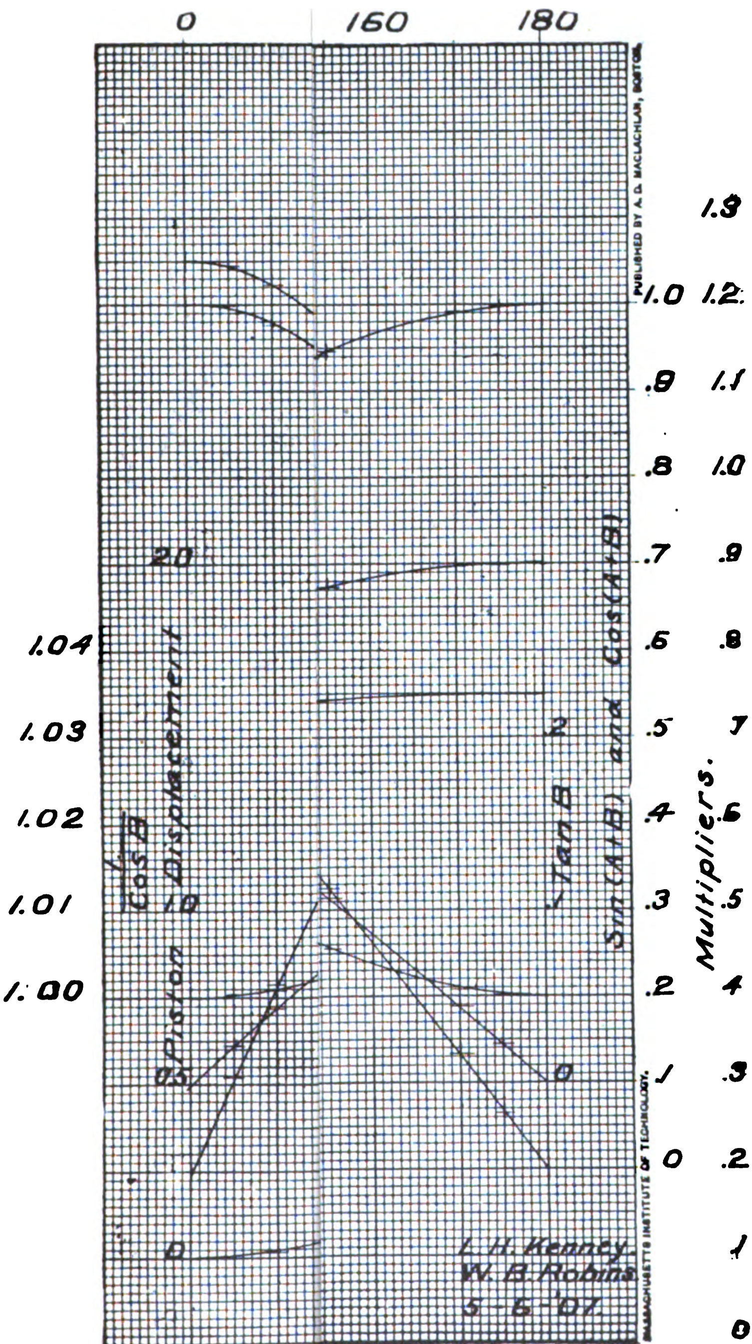
Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Main Bearing No. 1:													
$\frac{P}{2}$ F.L.P. down	—72 250	—58 250	—41 000	—11 650	21 575	45 300	59 300	71 800	83 100	85 050	81 350	74 200	66 250
$\frac{P}{2}$ F.L.P. up	—77 700	—88 750	—88 300	—79 850	—59 600	—38 950	—26 600	—16 600	0	10 650	23 425	33 650	45 750
s	—37 750	—37 150	—34 200	—26 700	—15 970	—6 560	0	6 560	15 970	26 700	34 200	37 150	37 750
S	—34 320	—33 720	—30 770	—23 270	—12 540	—3 130	3 430	9 990	19 400	30 130	37 630	40 580	41 180
$\frac{S}{2}$	—17 160	—16 860	—15 385	—11 635	—6 270	—1 565	1 715	4 995	9 700	15 065	18 815	20 290	20 590
Valve gear, F.L.P. down	14 200	15 000	15 500	14 700	12 800	10 500	8 700	6 800	3 600	—900	—5 300	—8 100	—9 700
Valve gear, F.L.P. up	14 200	13 000	10 700	6 700	1 600	—2 000	—4 700	—6 800	—9 400	11 300	—11 600	—10 600	—9 700
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, down	—75 210	—60 110	—40 890	—8 585	28 110	54 240	69 720	83 600	96 400	99 220	94 870	86 390	77 140
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, up	—80 660	—92 610	—92 990	—84 790	—64 270	—42 520	—29 590	—18 410	300	14 420	30 640	43 340	56 640
Pressure per square inch of bearing, down	—179.5	—143.5	—97.6	—20.5	66.1	129.5	166.3	199.6	230.0	236.8	226.5	206.2	184.0
Pressure per square inch of bearing, up	—192.6	—221.0	—222.0	—202.5	—153.2	—101.5	—70.6	—43 9	0.7	34.4	73.1	103.5	135.2
Main Bearing No. 2:													
$\frac{P}{2}$ H.P. $\frac{P}{2}$ F.L.P. + S down	21 580	33 760	41 980	50 180	62 180	65 380	68 180	66 880	57 330	39 980	19 830	3 780	—5 570
$\frac{P}{2}$ H.P. $\frac{P}{2}$ F.L.P. + S up	8 280	—4 870	—20 220	—16 520	1 780	10 980	9 580	4 980	—14 425	—37 570	—60 820	—70 870	—68 820
Pressure per square inch of bearing, down	52 5	82.1	102.2	122.0	151.3	159.1	166.0	162.8	139.3	97.3	48.3	9.2	—13.6
Pressure per square inch of bearing, up	20.2	—11.8	—49.2	—40.2	4.3	26.7	23 3	12.1	—35.1	—91.4	—148.0	—172.5	—167.4
Main Bearing No. 3:													
$\frac{P}{2}$ H.P. down	- 27 600	—3 325	15 125	36 100	58 750	78 550	91 350	102 400	113 500	116 400	104 700	89 100	68 700
$\frac{P}{2}$ H.P. up	—61 400	—82 500	—105 000	—105 000	—84 750	—64 250	—53 150	—43 750	—36 000	—79 350	—23 250	—16 050	0
Valve gear down	—4 000	—4 400	—4 800	—4 100	—2 700	—1 000	300	1 800	4 300	7 400	9 800	11 100	11 800
Valve gear up	—4 000	—3 200	—1 800	400	3 400	5 600	6 900	8 200	9 800	11 500	12 200	12 200	11 800
$\frac{P}{2}$ $\frac{S}{2}$ + valve gear, total, down	—48 760	- 24 590	—5 060	20 370	49 780	75 990	93 370	109 200	127 500	138 900	133 300	120 500	110 100
$\frac{P}{2}$ $\frac{S}{2}$ + valve gear, total, up	—82 560	—102 600	—122 190	—116 200	—87 620	—60 220	—44 540	—30 560	—16 500	—2 785	7 765	16 440	32 390
Pressure per square inch of bearing, down	—116.3	—58.7	—12.1	48.6	118.9	181.5	223.0	260.3	304.0	331.5	318.0	287.4	262.5
Pressure per square inch of bearing, up	—197.0	—244.7	—291.3	—277.0	—209.0	—143.7	—106.4	—72.9	—39.4	—6.7	18.5	39.2	77.3

MAIN BEARING PRESSURES—Continued.

Downward forces are +
Upward forces are —

Pressures in pounds.
Projected bearing area is used.

Main Bearing No. 4:													
$\frac{P}{2}$ I.P. { do wn.	—21 775	—4 665	10 290	34 850	53 800	72 000	87 100	99 700	114 000	118 650	101 250	81 500	61 750
$\frac{P}{2}$ I.P. { up.	—60 650	—77 750	—90 200	—98 000	—91 000	—71 600	—57 600	—43 550	—28 000	—21 000	—15 550	—4 665	20 575
Valve gear, { down.	14 000	14 700	15 300	14 800	12 800	10 900	9 400	7 900	5 200	1 200	—2 600	—5 300	—6 700
Valve gear, { up	14 000	12 800	10 700	6 900	2 500	—700	—2 700	—4 500	—6 700	—8 200	—8 500	—7 700	—6 700
$\frac{P}{2}+\frac{S}{2}+$ valve gear, total, { down	—24 940	—6 825	10 210	38 020	60 330	81 340	98 220	112 600	128 900	134 900	117 500	96 490	75 640
$\frac{P}{2}+\frac{S}{2}+$ valve gear, total, { up	—63 810	—81 810	—94 890	—102 700	—94 770	—73 870	—58 590	—43 060	—25 000	—14 140	—5 235	7 925	34 470
Pressure per square inch of bearing, { down	—59.5	—16.3	24.4	90.7	143.9	194.0	234.1	268.7	307.5	321.8	280.3	230.3	180.5
Pressure per square inch of bearing, { up	—152.2	—195.3	—226.3	—245.0	—226.2	—176.4	—139.8	—102.8	—59.6	—33.7	—12.5	18.9	82.2
Main Bearing No. 5:													
$\frac{P}{2}$ I.P. + $\frac{P}{2}$ A.L.P. + S { down	19 960	27 070	33 730	47 860	54 040	58 400	62 230	63 730	61 030	46 530	19 530	—1 320	—14 670
$\frac{P}{2}$ I.P. + $\frac{P}{2}$ A.L.P. + S { up	4 280	—2 520	—7 370	—9 520	—5 970	2 580	6 030	7 380	1 130	—24 050	—48 570	—55 190	—40 750
Pressure per square inch of bearing, { down	31.0	42.1	52.5	74.5	84.1	90.8	96.8	99.1	95.0	72.4	30.4	—2.1	—22.7
Pressure per square inch of bearing, { up	6.7	—3.9	—11.5	—14.8	—9.3	4.0	9.4	11.5	1.8	—37.4	—75.5	—85.8	—63.4
Main Bearing No. 6:													
$\frac{P}{2}$ A.L.P. { down	—64 750	—53 950	—36 450	—6 475	25 700	47 500	60 200	70 750	81 600	85 050	79 400	71 800	61 500
$\frac{P}{2}$ A.L.P. { up	—79 850	—86 250	—85 150	—75 550	—56 400	—39 400	—28 300	—17 035	—3 190	9 580	20 010	28 300	38 300
Valve gear { down	14 500	15 300	15 700	14 900	12 800	10 600	8 800	6 900	3 800	—700	—5 200	—8 000	—9 400
Valve gear { up	14 500	13 300	10 800	6 800	1 900	—2 000	—4 700	—6 700	—9 300	—11 300	—11 500	—10 500	—9 400
$\frac{P}{2}+\frac{S}{2}+$ valve gear, total, { down	—67 410	—55 510	—36 140	—3 100	32 230	56 340	70 720	82 650	95 100	99 420	93 020	84 090	72 690
$\frac{P}{2}+\frac{S}{2}+$ valve gear, total, { up	—82 510	—89 810	—89 740	—80 390	—60 770	—42 970	—31 290	—18 740	—2 790	13 350	27 330	38 090	49 490
Pressure per square inch of bearing, { down	—161.0	—132.5	—86.2	—7 7	76.9	135.0	168.8	197.3	227.0	237.2	222.0	201.0	173.5
Pressure per square inch of bearing, { up	—197.0	—214.5	—214.2	—192.0	—145.1	—102.5	—74.7	—44.7	—6.7	31 9	65.2	90.9	118.1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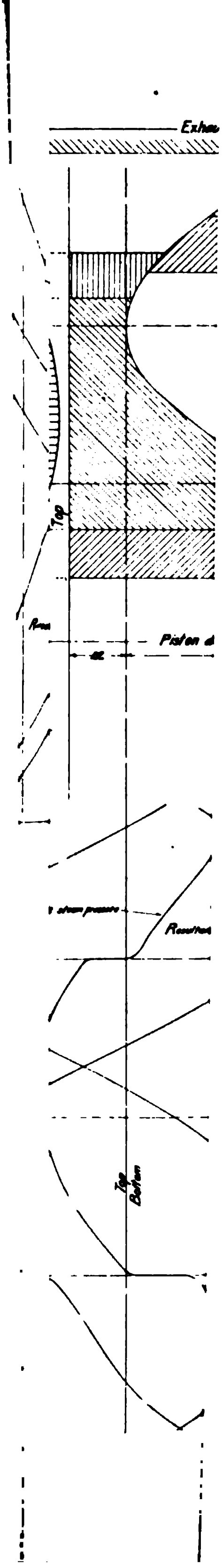
5-6-'07
PUBLISHED BY A. D. MACLACHLAN, BOSTON
MASSACHUSETTS INSTITUTE OF TECHNOLOGY.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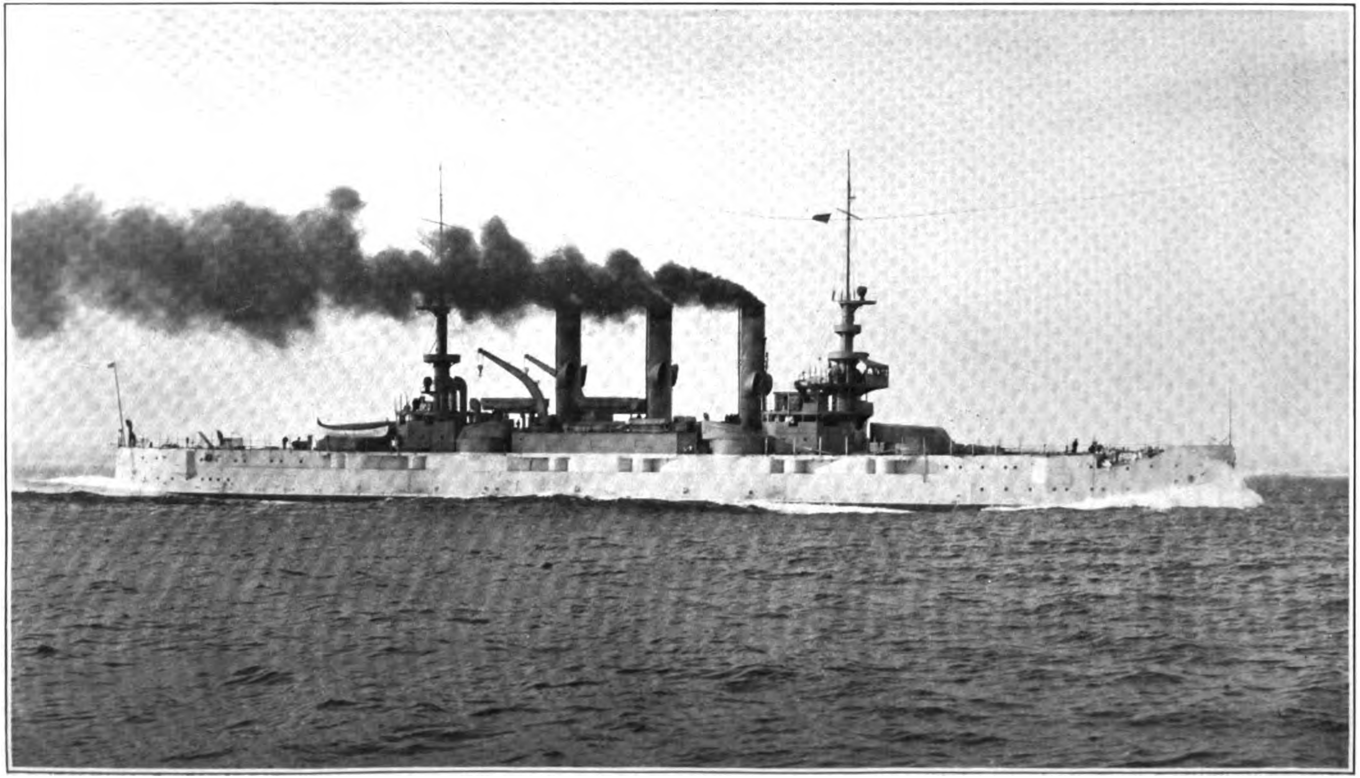


U. S. S. "CONNECTICUT."

U. S. S. *CONNECTICUT*.

OFFICIAL TRIAL.

The *Connecticut* is one of two battleships authorized by Act of Congress, July 1st, 1902, the sister ship being the *Louisiana*, built by the Newport News Shipbuilding & Dry Dock Company, at Newport News, Va. For a full description of this latter vessel, which applies equally to the *Connecticut*, see Volume XVIII, page 171.

STANDARDIZATION TRIAL.

The standardization runs were made over the measured-mile course off Rockland, Me., August 7th, 1907. The weather was cloudy to clear, with a light breeze from South. Smooth sea.

Previous to the vessel starting on the standardization runs, the draught was taken and found to be—

Forward, feet and inches,	24–00
Aft, feet and inches,	26–01
Mean, feet and inches,	25–00½
Corresponding displacement, tons,	16,440

These runs gave the data from which the curves on Plate I were plotted.

The results of the runs with and against the tide having been plotted as separate curves, the curve of true speed was obtained, from which it was deduced, from a mean of both engines, that 120.3 revolutions per minute would be required for a true speed of 18 knots.

After the standardization trial was completed, and the vessel was again at anchor, the draught was found to be—

Forward, feet and inches,	23–09½
Aft, feet and inches,	26–00
Mean, feet and inches,	24–10¾
Corresponding displacement, tons,	16,300

FOUR-HOURS' FULL-POWER TRIAL.

The four-hours' full-power trial was held on August 9, 1907. The weather was clear with a light northerly breeze. Smooth sea.

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*]

	Starboard.	Port.
Mean steam pressure at boilers, pounds	263.5	
Mean steam pressure at engines, pounds		
H.P. steam-chest, gauge	224.0	228.0
1st receiver (absolute), pounds	106.0	108.0
2d receiver (absolute), pounds	37.0	29.6
Vacuum in condensers, inches of mercury, mean	21.0	20.9

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Injection, degrees	63	63
Discharge, degrees	112	118
Hotwell, degrees	90	90
Feed water, degrees	152	153
Engine room, working platform, degrees	87	94
Firerooms, working level, degrees	93	
Smoke stacks, average, degrees	571	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	Port.
Average revolutions, main engines, per minute	128.48	128.04
Mean revolutions, both engines, per minute	128.26	
Pumps, main air	25.0	26.8
circulating	266.5	211.5
feed, d.s., per minute, aft	37.4	35.0
forward	42.3	38.8
fire and bilge	29.3	
Auxiliary condenser, air and circulating	42.3	20.4
Blower engines	543.5	
Speed of ship, in knots per hour	18.783	
Slip of propeller, in per cent. of its own speed, based on mean pitch	15.94	15.76
Air pressure in firerooms, in inches of water, mean	1.36	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

	Starboard.	Port.
Main engines, H.P. cylinder	82.38	85.61
I.P. cylinder	43.53	51.92
F.L.P. cylinder	26.68	23.71
A.L.P. cylinder	24.62	21.51
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	53.53	53.59

INDICATED HORSEPOWER.

		Starboard.	*Port.*
Main engines,	H.P. cylinder	2,080	2,149
	I.P. cylinder	2,963	3,523
	F.L.P. cylinder	2,412	2,136
	A.L.P. cylinder	2,234	1,836
	total	9,689	9,644
Collective H.P. of both main engines		19,333	
Air, circulating, feed and hotwell pumps		486	
All other auxiliaries in operation		706	
Total all machinery		20,525	

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	18.74
Cooling surface (main condenser), square feet per I.H.P. (main engines)	1.072
Heating surface, square feet per I.H.P. (total)	2.57

NOTES.

BABCOCK & WILCOX BOILERS IN THE ROYAL NAVY.

The first of H. M. vessels to be supplied with Babcock & Wilcox patent water-tube boilers was the torpedo gunboat *Sheldrake*, having four boilers, each two of which were placed back to back in separate compartments, and fired fore and aft.

These boilers were made under rigorous survey by Admiralty inspectors, in accordance with the terms of the contract, and one of the four was erected at the constructors' works and there subjected to a number of tests to determine its efficiency.

TRIAL OF H. M. S. *SHELDRAKE*.

At the expiration of the usual basin and sea-going trials the Admiralty decided to test the boilers under actual sea-going conditions, and accordingly an exhaustive series of trials—nine in number—were carried out, each over a distance of 1,000 miles.

The results obtained on these, and the preceding basin and commissioning trials, were as follows:

Date.	Nature of trial.	No. of boilers in use.	No. of boilers in reserve.	Duration in hours.	Steam pressure.	Indicated horsepower.	Air pressure in stokehold.	Coal per I.H.P., all purposes.	Coal per square foot grate.	Water per pound coal.	Ashes per pound coal.	Grate surface.
14-11-98	Evaporative	2	2	8	168	1,116	.0	1.69	15.0	10.5	...	126
15-11-98	Evaporative	2	2	8	179	1,292	.0	1.46	15.0	10.5	...	126
16-11-98	Evaporative	2	2	8	169	1,761	...	1.78	25.0	9.05	...	126
17-11-98	Evaporative	2	2	8	165	1,873	...	1.67	25.0	9.34	...	126
28-11-98	8 hours at 2,500 I.H.P	4	...	8	152	2,742	.0	1.43	15.0	...	...	252
1-12-98	3 hours at 3,000 I.H.P	4	...	3	151	4,050	.43	1.57	25.6	...	...	252
22- 2-99	3 hours commissioning	4	...	3	119	2.735	.14	1.64	17.8	...	...	252
28- 2-99	1,000 miles at 1,500 I.H.P	3	1	69	120	1,301	.0	1.61	12.8	...	.152	189
9- 3-99	1,000 miles at 1,500 I.H.P	...	1	68	120	1,506	.0	1 6	12.67	...	.150	189
28- 3-99	1,000 miles at 1,500 I.H.P	3	1	70	135	1,534	.0	1.75	14.2	...	.200	189
20- 4-99	1,000 miles at 1,500 I.H.P	3	1	68½	130	1,539	.c	1.59	13 1	...	.150	189
5- 5-99	1,000 miles at 1,800 I.H.P	3	1	67	135	1,829	.0	1.6	15.4	...	.216	189
19- 5-99	1,000 miles at 1,800 I.H.P	3	1	66½	140	1,838	.0	1.68	16.4	...	.220	189
15- 6-99	1,000 miles at 2,000 I.H.P	3	1	59	145	2,033	.0	1.57	17.0	...	.220	189
3- 7-99	1,000 miles at 2,000 I.H.P	3	1	61¼	140	2,042	0	1.56	16.8	...	.230	189
20- 7-99	1,000 miles at 2,250 I.H.P	3	1	56¾	150	2,245	.0	1.63	19.4	...	.250	189
Total		...	...	632¾	...	...	...	...	...	...	...	...
Mean		...	...	...	146	1,974	...	1.63	...	9.85	.198	...

SUMMARY—BASIN AND SEA-GOING TRIALS OF H. M. S. *SHELDRAKE.*

At the finish of the 1,000-mile trials, several experiments were made. The first one was on the forward boiler, when the time required to raise steam to 140 pounds pressure from cold water was taken; the temperature of the water at the start was 70 degrees, and steam was raised to 140 pounds pressure in 23 minutes. After that a stopping and starting test was made; the engines were going full speed and suddenly stopped. The front tube doors and uptake doors were immediately opened and the ash-pit doors closed; the steam gauge was then watched, and the pressure did not rise more than 5 pounds, neither did the safety valves lift.

The nxet test was made to ascertain how soon the operation of drawing a tube could be commenced after the fires had been pulled out of the furnace. No. 4 boiler was used for this purpose. This boiler was worked at full power; suddenly the fires were drawn and the water blown out; in 24 minutes after hauling the fires several caps were taken off ready for drawing tubes. Then a test was made to show how quickly a tube could be taken out of a boiler. Three tubes were drawn, one after the other; the first took 11 minutes, the second 10 minutes, and the third 9 minutes.

At the conclusion of these experiments, the *Sheldrake* had completed the whole of the Admiralty program and returned to Devonport, where a careful examination was made of the boilers, which were found to be in as good a condition everywhere as when they left the works.

Since the *Sheldrake* trials were carried out the following vessels in H. M. Navy have been fitted, or are about to be fitted, with these boilers:

Name.		Indicated horsepower	Year.	No. of Babcock & Wilcox boilers.
Sloop	*Espiegle*	1,400	1899	4
Sloop	*Odin*	1,400	1900	4
Cruiser	*Challenger*	12,500	1900	12
Cruiser	*Hermes*	10,000	1901	12
Battleship	*Queen*	15,000	1901	15
Cruiser	*Cornwall*	22,000	1901	24
Battleship	*Dominion*	18,000	1902	16
Battleship	*Commonwealth*	18,000	1902	16
Battleship	*King Edward VII*	18,000	1902	10
Cruiser	*Argyll*	21,000	1902	16
Battleship	*Hindustan*	18,000	1902	18
Cruiser	*Black Prince*	23,500	1903	20
Cruiser	*Duke of Edinburgh*	23,500	1903	20
Battleship	*Britannia*	18,000	1904	18
Battleship	*Hibernia*	18,000	1904	18
Battleship	*Africa*	18,000	1904	18
Battleship	*Lord Nelson*	16,750	1904	15
Cruiser	*Minotaur*	27,000	1904	25
Battleship	*Dreadnought*	23,000	1905	18
Cruiser	*Indomitable*	41,000	1905	31
Battleship	*Bellerophon*	23,000	1906	18
Battleship	*Superb*	23,000	1906	18

The greater number of these vessels have already successfully passed their official trials, and some very interesting particulars of these are to hand from Messrs. Babcock & Wilcox, to whose courtesy we owe the accompanying illustrations.

In connection with the following details of trials it should be noted in every case that the fuel consumed is for all purposes, but the indicated horsepower is that of the main engines only.

H. M. S. *DREADNOUGHT.*

H. M. S. Dreadnought has eighteen Babcock & Wilcox boilers installed, having a total tube-heating surface of 55,530 square feet, and a total grate area of 1,599 square feet. Her very satisfactory trials are a matter of recent history. The evaporation at all trials worked out to between 10 to 10.2 pounds per pound of coal per hour, which, under the service conditions prevailing during the trials, must be regarded as an excellent result. During the course of the week's steaming it was decided to ascertain the maximum power which could be realized, and 28,000 indicated horsepower was developed ac-

cording to the torsion meter. This gives an overload, a power above the maximum provided for in the design, of 5,000 horsepower; the coal consumption with this overload equalled about 1.4 pounds per shaft horsepower per hour.

H. M. S. *KING EDWARD VII.*

H. M. S. King Edward VII has ten Babcock & Wilcox boilers, having a total tube-heating surface of 27,620 square feet, and a total grate area of 815.5 square feet; and six cylindrical boilers, having a total heating surface of 16,320 square feet, and a total grate area of 491 square feet.

RESULT OF OFFICIAL TRIALS, H. M. S. *KING EDWARD VII.*

......	Low power.	Maximum continuous.	Full power.
Date of trial	Nov. 16th and 17th, 1904.	Nov. 18th and 19th, 1904.	Nov. 23d, 1904.
Duration of trial, hours	30.0	30.0	8.0
Number of boilers in use	6 Cylindrical	6 Cylindrical and 10 B. & W.	6 Cylindrical and 10 B. & W.
Heating surface, square feet	16,320.0	43,940.0	43,940.0
Grate area, square feet	491.0	1,306.5	1,306.5
Fuel, kind used	Welsh coal.	Welsh coal.	Welsh coal.
Steam, average observed gauge pressure, pounds per square inch	193.0	194.0	200.0
Draft, inches water pressure in stokehold	Nil.	Nil.	B. & W., 0.55 Cyl., 0.71
Indicated horsepower	3,760.0	12,844.0	18,138.0
Indicated horsepower per square foot of grate	7.66	9.83	13.88
Coal, total consumed per hour, pounds	9,889.0	25,174.0	39,359.0
Coal per indicated horsepower per hour, pounds	2.63	1.96	2.17
Coal per square foot of fire-bar surface per hour, pounds	20.1	19.2	30.0
Heating surface per indicated horsepower	4.34	3.42	2.42
Revolutions per minute	69.8	106.1	116.5

COMPARATIVE TRIALS TO OBTAIN RELATIVE COAL CONSUMPTION OF BABCOCK & WILCOX AND CYLINDRICAL BOILERS.

The official trials of this vessel, which is fitted with two-fifths cylindrical boilers and three-fifths Babcock & Wilcox boilers, not having been so satisfactory as those of similar vessels fitted entirely with Babcock & Wilcox boilers, a series of comparative trials were ordered to be carried out, using alternatively cylindrical boilers alone and Babcock & Wilcox boilers alone.

Before this was commenced, however, numerous alterations were made to the cylindrical boilers with the object, if

possible, of improving their performance, and they were fitted with retarders and generally in accordance with the most modern practice in all respects.

The conditions were that four trials were to be made, each of eight hours' duration. On two of these trials all the cylindrical boilers were to be used, and on the other two, a sufficient number of Babcock & Wilcox boilers were to be used to give the same proportion of the total power as the cylindrical boilers. The total power of the ship is 18,000 indicated horsepower. The cylindrical boilers were designed to give two-fifths of this power—namely, 7,200 indicated horsepower; and the proportion of the Babcock & Wilcox boilers used on the trials, was to be capable of giving the same proportion of the total power—viz: two-fifths—7,200 indicated horsepower.

On the first pair of trials each set of boilers was to develop 3,600 indicated horsepower.

COAL PER INDICATED HORSEPOWER.

On the second pair of trials the cylindrical boilers were to develop the maximum indicated horsepower of which they were capable, and then the Babcock & Wilcox boilers were to burn the same aggregate quantity of coal in order to ascertain what power could be obtained from them.

The results conclusively proved the superiority of the Babcock & Wilcox boilers, and were as follows:

......	One-fifth power trial.		Full power trial.	
	Cylindrical.	Babcock & Wilcox.	Cylindrical.	Babcock & Wilcox.
Indicated horsepower..............	3,634.0	3,759.0	6,686.0	7,510.0
Coal per indicated horsepower.	1.8	1.74	1.88	1.67

It should be noted that the cylindrical boilers failed to obtain their designed power—viz: 7,200; whereas the Babcock & Wilcox boilers easily obtained it, and they could have developed a much larger power if required, their limitation being the total quantity of coal, which was not permitted to exceed that burnt in the cylindrical boilers.

......	Low power.	Maximum continuous.	Full power.
Date of trial	Oct. 3d and 4th, 1906.	Oct. 6th and 7th, 1906.	Oct. 9th, 1906.
Duration of trial, hours	30.0	30.0	8.0
Number of boilers in use	9	18	18
Heating surface, square feet	27,765.0	55,530.0	55,530.0
Grate area, square feet	799½	1,599.0	1,599 0
Fuel, kind used	Welsh coal.	Welsh coal.	Welsh coal.
Steam, average observed gauge pressure, pounds per square inch	220.0	230.0	241.0
Draft, inches water pressure in stokehold	0.56	0.9	1.2
Indicated horsepower	5,013.0	16,930.0	24,712.0
Indicated horsepower per square foot of grate	6.27	10.58	15.45
Coal, total consumed per hour, pounds	12,955.0	28,815.0	37,315.0
Coal per indicated horsepower per hour, pounds	2.58	1.7	1.51
Coal per square foot of fire-bar surface per hour, pounds	16.2	18.02	23.33
Heating surface per indicated horsepower.	5.54	3.28	2.25

OFFICIAL TRIALS OF H. M. S. *DREADNOUGHT.*

The first-class battleship *Britannia,* of 18,000 indicated horsepower, is installed with 18 Babcock & Wilcox boilers, having a total tube heating surface of 40,020 square feet, and a total grate area of 1,250 square feet, and three cylindrical boilers, having a total heating surface of 8,100 square feet, and a total grate area of 247 square feet. Six of the 18 Babcock & Wilcox boilers are fitted with their patent superheaters.

THE NAVY REPAIRS QUESTION.

In the last Navy debate in Parliament Mr. Balfour pointed out that, so far as he knew, the whole disquiet of the public mind regarding the position of the Navy related to repairs, and in writing on this debate in a recent issue we promised to investigate this question. There is, as Mr. Balfour said, some disquiet in the public mind, and it is possible for the superficial critic to make some sort of a case against the Admiralty because of the number of ships awaiting repairs, each with a more or less important list of defects. But a closer examination of the whole situation, and an analysis of the comparative fighting efficiency of the ships on the list, reduces to vanishing point what is referred to by some writers as a serious situation.

Most of the ships, indeed, belong to the second line, and many of the defects do not concern the fighting machinery. For this and other reasons the Admiralty policy, while ensuring permanent economy, does not affect the preparedness of the Fleet to maintain our supremacy at sea.

The attention devoted to the repairs question is indeed an evidence of that public interest in the Fleet which is an important factor in the maintenance of its efficiency; the outcry is itself proof of this general efficiency. Under the conditions antecedent to 1903 a ship, after she had been in commission three years, and in some cases four years, returned to the dockyard, to be passed into the Dockyard Reserve. If it was not convenient to undertake the refitting or repairs necessary, she was quietly forgotten. Even when refitted and passed into the Fleet Reserve, she remained there without anyone being directly responsible for her maintenance in a satisfactory condition. A company of officers and men, it is true, boarded her from time to time; but as they were seldom called upon to go to sea with the ship, they had little direct interest in the ship or her machinery. The consequence of this inattention, if not neglect, was that when the vessel went on maneuvers with a more or less scratch screw, her performances were unavoidably unsatisfactory. Under present conditions a ship is expected, after she enters the Fleet new, to be in active service practically continuously until she becomes obsolescent. She is commissioned for two years. At the end of this time, if no extensive repairs are required, she is re-commissioned. If, however, changes are necessary, either as the result of wear and tear, or of current improvements in mechanical appliances, the vessel is passed into the Home Fleet, there to await a convenient opportunity for undertaking such moderate refit as is necessary. If she goes to the Nore Division, she has a full crew; if to the other divisions, a nucleus crew. If the refit required is extensive, the vessel is placed in dockyard hands and a temporary crew is appointed for care and maintenance purposes and to assist the dockyard in the repair work. The consequence is that at no time in the history of the ship is she with-

out a staff who is responsible for her efficiency. Thus, when repairs are called for, and the non-fulfillment of them, or delay in undertaking them, militates against efficiency, the officers naturally make it their business to enlist official and, in some cases, public influence to ensure that the repairs will be carried out. In the past there was the possibility of neglect without disclosure. Today there may be neglect, but the responsibility for non-efficiency comes home so directly that there must be disclosure. This is all for the good of the Service. Indeed, it is the duty of the commander-in-chief of each fleet to report periodically direct to the Board of Admiralty as to the state of his ships, and immediately in the event of any defect being discovered; so that here, again, there is advantage in having so many ships in commission, even if the crew be of nucleus strength.

Such disclosure of delay may be an explanation of the increased attention directed to the subject, as the extent of delay does not justify the outcry: the delay is not nearly so serious as in the past. It is difficult to measure the repair requirements of the Fleet. In industrial concerns, and in merchant fleets, experience has established a certain rate per cent. per annum for repairs; but there are very few concerns where this could be justified as accurate year by year; it is rather an arbitrary figure accepted for accounting. The case of the Fleet is still more difficult. The repairs of the Fleet depend on many factors—on the amount of work done by the ships and on the number of accidents partly consequent upon officers taking risks in maneuvers which would be inevitably forced upon them in war. There is also the question of the advance in invention, which may make small or extensive changes imperative. There is, for instance, the present case of supplying cold air for the effective ventilation of magazines, the need of which was established by research in the deterioration of explosives under high temperature. Equally operative is the age of the ships. The scrapping of several obsolescent ships a year or two ago necessarily reduced the repairs bill to a large extent. Finally, there is the amount of work undertaken by the ship's

company; and in this respect great advances, as will presently be explained, have been made in British warships. But notwithstanding this, from £1,500,000 to £2,000,000 are spent per annum; during the past five years £11,000,000 have been spent on repair work.

When, under the present régime, a scheme was elaborated in order to ensure efficiency in every department of the naval service, the organization equipment, and general system of repair work at the dockyards were included, and it was decided to discontinue the practice of giving out repair work to contract. At the same time it was considered desirable to reduce to the minimum the amount of new constructional work undertaken at the dockyards. As it reduced the probabilities of extensive refits the scrapping of a large number of old ships involved a reduction in the number of workmen employed at the dockyards. This was commendable, as experience had shown that there was lack of economy, because the number of workers could not be varied to suit the great fluctuations from time to time in the amount of repair work to be done. It was not possible to vary the number of men employed in the dockyards as in the Clyde district, for instance, where the opportunities for re-engagement in another establishment are so great that the men discharged seldom experience hardship. Under the old conditions the whole of the squadron returned at one time for refit, and there was a glut of work; at other times there was a paucity. The machinery of a ship was invariably opened up. If the defects discovered were extensive, and there was otherwise plenty of work, the ship was passed to the reserve, to be repaired at a more convenient season. Cylinders, bearings, &c., were as often as not left open, so that there was serious deterioration. Moreover, when there was little work to do, there was a tendency to create work, and machinery might be opened up in order to discover whether a job were possible, without any apparent reason, and thus money was wasted without the efficiency of the Service being improved, and simply to obviate the discharge of men. There is now no opening up of machinery and boilers by the dock-

yard, unless the engineer of the ship has reason to believe that such is desirable.

Instead of the whole of a squadron returning simultaneously for refit and repair, not more than three battleships and one armored cruiser can be detached from each fleet at any time for defects to be made good. Here we have a very important advantage in maintaining the preparedness of the Fleet for action. By this system there cannot be repeated the serious condition which prevailed at the time of the Dogger Bank incident, when the whole of the vessels of the cruiser squadron were undergoing repairs.

The number of men employed has been reduced to the normal staff considered suitable to meet the normal volume of repair and new constructional work. Great fluctuations are avoided without abnormal delay in refit work, undue creation of defects, or prolongation of work. If the number of workers is sufficiently large for well-organized repair work there is a certainty of a better efficiency. The whole question resolves itself into one of determining the urgency of repairs from the point of view of the efficiency of the Fleet and its preparedness for action. In this the Admiralty are only following private practice, which is based on sound commercial experience. It is true, of course, that the Navy must be maintained in efficiency, apart altogether from financial considerations; but that does not preclude the authorities from combining economy with efficiency. This latter term, too, is relative. If we were weak in any of the types of warships in comparison with probable opponents, it would be necessary to ensure that every vessel was at all times available for service; but the view is now accepted that our Fleet is at least numerically superior to the two-Power standard. It therefore becomes in part a question as to the relative number of vessels on the repair list in our own Navy and that of foreign Powers. We are satisfied that in this respect we are in an advantageous position under the present system. Consequently there must be approval of the principle now adopted of considering the importance of the defects of one ship against another, and of weighing the value

of each ship in maintaining the composition of our sea-going fleets. Thus the *Dreadnought* or the *King Edward VII* must have preference to a vessel of the *Royal Sovereign*, *Majestic* or *Canopus* classes. The same applies to cruisers. Again, it is incumbent upon the authorities to give preference to such repairs in any one ship as affect the fighting power of that ship. Thus it would be more important to replace the controlling valve of a hydraulic gun mounting than to repair the mechanism of an ash hoist. Both are important, but in very different degree. The urgency with which the work should be undertaken must vary in direct ratio to the influence which the defect has upon the fighting efficiency of the ship or of the Fleet.

Another consideration is that by concentrating, as far as necessary, attention upon the repairing of ships of high fighting efficiency, the preparedness of the Fleet is maintained at a higher standard, and the training of the officers and men made more continuous. When the active-service ratings are not sufficient to completely man all the vessels in the Fleet, it is better that they should man all the ships that are immediately ready to fight, leaving the remainder partially empty, for the dockyards to complete at high pressure when the contingency arises. The ships in reserve awaiting repairs may be manned by less efficient men from the Fleet or the Royal Naval Reserve. This provides a reserve of ships, which, in the event of war, would very quickly be brought to a state of efficiency by an emergency increase in the staff of the dockyards.

There seems no present reason for maintaining the dockyard staff at an abnormally high standard when there is not continuous work for them. If experience proves that under the present conditions in the dockyards it is not possible to obviate a permanent delay in the repair of the ships, then it will be easy to increase the numbers; but it has been demonstrated in the past that it is easier to engage than to get rid of men in the dockyards, and that surplus labor, in view of the absence of continuity in the volume of repairs, militates seriously against economy. That the staff is adequate has

been proved by the great increase in rapidity with which repair work is now carried out. Formerly a thorough refit for a battleship occupied anything from twenty to forty months, whereas now, without any overtime, six months suffice. Dilatory work is always expensive, and thus the rapid completion of repair work not only ensures the return of the unit to the fighting line in a much shorter period of time, but also makes for economy.

An important factor in the problem has reference to the amount of work done on board ship for the maintenance of machinery in a thorough state of repair. Now that each ship is continuously in commission, instead of being passed into the Fleet or Dockyard Reserves, officers naturally gain credit for the maintenance of their ship in a satisfactory condition, and are, we hope, commended for undertaking important work for the repair of their machinery. Every ship of a size larger than a destroyer has now a very extensive workshop. In a battleship, for instance, there are fitted power-driven horizontal boring machines, screw-cutting lathes, ranging up to fairly respectable capacity, a side-planing machine, various drilling machines, punching and shearing machines, tool-sharpening, twist-drill grinding, and other tools. There is a respectable sized forge, with furnaces and appliances for producing small castings. Some of the engineer officers have done very good work to meet emergencies. In one case that we know of a brass casting of 116 pounds weight was made with the appliances on board. The ships carry rough castings and unmanufactured articles, the use of which for renewal of parts can be anticipated. On occasion the engine-room artificer ratings are relieved of watch-keeping duties for this repair work. From several engineer officers we have heard of the enthusiasm of the artificers in this return to the work of their earlier years; they have often continued to work when they were supposed to be off watch. In every complement, too, there are usually found representatives of almost every trade associated with the manufacture of machinery, and the result is a spirit of emulation between men of each ship and between ships, which is most conducive to the efficiency of the Service.

Repair ships are now attached to the principal fleets, and these are to all intents and purposes floating factories, having a pattern shop, joiner's shop, foundry, smithy, boiler shop, coppersmith's and plumber's shops, and a machine and fitting shop for engines, boiler and electrical work. The forge and plate furnaces are usually run by motor fans. There are the usual slabs for bending plates, angles and bars. The foundries can undertake pretty heavy castings, and are fitted with electric cranes. The coppersmith's shop has the usual forges, slabs, and bending machines, &c. The carpenter's shop has circular and other machine saws and tools. As a consequence the ships do not require to come back to the dockyard so frequently for repairs.

It will be realized that the repair work has been thoroughly systematized as an important part of naval re-organization, and will, as in other departments, be conducive to the improvement of naval efficiency. We are convinced that the scheme will eventually effect a great economy without militating in the slightest degree against preparedness for war.—"Engineering."

"BACK FLASH" FROM MODERN SMOKLESS POWDER.

The recent terrible disaster on board of the Japanese battleship *Kashima,* in which over forty officers and men were killed and injured owing to charges of modified cordite, while waiting to be loaded into a 10-inch gun, being ignited by the "back-flash" from the previous charge just fired, once more directs serious attention to an ever-present danger which threatens our own Navy. Both nations use practically the same pattern of gun and absolutely identically the same "powder"—or, to be more precise, the same propellant—namely, modified cordite. The tale of these accidents is mounting up; the United States has to deplore the loss of many of her sailors from exactly the same cause, though the propellant used in the United States Navy is of a somewhat different type.

Briefly narrated, what happens is this: When a charge of

modern smokeless propellant is fired, the shot issues from the muzzle, but inside the bore of the gun there is left a large quantity of various mixed gases at a high temperature. Owing to their nature, however, they themselves are not sufficiently rich in oxygen to cause a visible flame. When the breech is opened there is an excess of fresh oxygen from the atmosphere, and the hot gases then burst into flame. If there is a draft down the bore of the gun from the muzzle to the breech, this flame is blown backwards a considerable distance to the rear of the gun, scorching and burning anything in its way. Should a charge of fresh propellant be ready behind the gun, waiting to be loaded after the shot is rammed home, it is not difficult to understand how there is certainty of terrible devastation, accentuated as it is by the small enclosed space in a turret, barbette or casemate, and the further possible danger of the flame spreading to charges in the ammunition hoists and thence down to the magazines.

As the velocity of the projectile is increased in each succeeding type of gun, the charges of propellant rendered necessary are greater, and thus the danger from back-flash is intensified, owing to the larger quantity of residual gases being left in the bore.

It was appreciated some time ago in our Navy that precautions must be taken to guard against such accidents as these, and the method since adopted is to expel the residual gases by an air-blast before the breech is opened; but this is at best a makeshift, and must unduly delay the service of the gun. Moreover, in the hurry of an action, sufficient time may not be given to the operation; nor if the ship is steaming to windward, or the guns pointing to windward, can it be safely depended upon to clear out all the gases.

The Germans seem to have solved the problem in a much more scientific and satisfactory manner by adopting for their Navy in 1906 a propellant powder which gives no back-flash, even in their largest guns. This has replaced in their Navy the modified cordite which is used at present in the British Navy. This final rejection, owing to the serious back-flash

defect, in favor of the flameless powder, known by the designation C/06, is a consequence of searching trials, extending over four years since 1902, the " /06" denoting the year in which it was adopted.

The exact composition of this powder is kept secret, but it consists of a mixture of nitro-glycerine and nitro-cellulose and vaseline, in which, so far, it closely resembles the composition of our modified cordite; but there is, in addition, a small quantity of some chemical which has the effect of preventing back-flash at the breech, and in smaller ordnance no flame is apparent at the muzzle. This latter is a considerable advantage during night firing, as the gun layer of the quick-firing weapon is not blinded by the intense brilliancy of the flame usual with our cordite, whereby the speed of firing is reduced and the accuracy of shooting impaired. It appears that the new German powder gives excellent shooting results.

So far as keeping qualities are concerned, the German powder is even better than our own cordite. This also is an important factor in view of the cordite explosions which have taken place from time to time in the magazines of British war vessels, fortunately, so far, without loss of life or of a ship; but similar good fortune may not continue. The report on these explosions has been kept extremely quiet, so that it is difficult to speak more fully of them; but on one occasion it was mere luck that a battleship was not sunk with all hands. Owing to this immunity from disaster, the public have not had their faith in the safety of ships in our fleet shaken in the same way as France quite lately was rudely awakened from her confidence in the excellence of the propellant powder used in her Navy by the terrible disaster on the *Jena*. Apart, however, from the question of instability of a powder, which can be met to a large extent by the installation of refrigerating machinery for the magazines, there remains the equally important question of doing away with back-flash. So far no mechanical means have been devised to entirely obviate the danger therefrom, and recourse must therefore be had to the chemist's art.

Germany has always been renowned for her progress in

chemistry, and in no branch has she shown herself more progressive than in matters connected with explosives. There is no single authenticated case of which we know of where an explosion has occurred on a German vessel of war, and the reason for this seems to lie in an intelligent anticipation of possible sources of danger and in the steps taken to prevent them. For example, as soon as nitro-smokeless powders replaced the smoke powders it was at once realized that the dictum of former days "keep your powder dry" must be changed to "keep your powder cool." All ships in the German Navy were fitted with apparatus to keep the magazines cool, but our Admiralty only seem to have appreciated the importance of this during the past year, when the matter was forced on the attention of the Government by the accident which occurred on H.M.S. *Fox*, though several years previously a similar accident had occurred on H.M.S. *Revenge*. In both cases, by a simple chance, the loss of the ships and their crews, fortunately, did not occur.

Now, again, it would appear that we shall have to follow the lead given by Germany and adopt a powder which will obviate the danger of back-flash in a proper and scientific manner. The grave fault in our administration appears to be that it is apparently no particular naval official's duty to watch the question of explosives and the progress made therein abroad. It is true that committees almost innumerable sit and report, but definite and prompt action does not appear to be taken upon the recommendations, or it is so leisurely that by the time the suggestions of the committee are put into effect they are practically out of date.

Committees themselves are not the most efficient methods of ensuring prompt reform. The members have frequently other duties, and are, moreover, not always suited by training and experience to properly consider the terms of reference. The result is that many of the earlier sittings are wasted in the elementary education of members in the subject under consideration. The sittings, too, are seldom, if ever, continuous; not more than an hour or so each week being devoted to effect-

ive investigation, and the period is still further broken up by holidays. Meanwhile much of the experimental work for the Committees is unnecessarily delayed. Finally, when the recommendations are made there is too often difference of opini due to a lack of full knowledge of the subject. The crux of the matter is therefore the concentration of responsibility by the appointment of an officer solely for explosives, whose duty it would be to acquire knowledge of foreign progress, maintain continuity in the records of British experience, and analyze the results of all operations in the Navy and in the private works devoted to research in explosive compounds. We are satisfied that with a fuller coordination between experimental work in private factories and naval battle practice, and with a firmer determination to overcome difficulties, our Admiralty would in this, as in other departments of naval activity, maintain our prestige.—"Engineering."

BATTLESHIP STRENGTH AND RELATIVE VALUE.

With October comes the intersessional political oratory, and this year we are promised a more than usually vigorous campaign. Two questions of interest to all connected with industry will be brought to the front: The one connected with trades-union organization and legislation, the other associated with naval strength and Admiralty administration. The former topic can only be considered when the demands of the labor socialist have been formulated. As regards the latter subject, the facts are indisputably in favor of the Admiralty policy during the past three years or so. We shall, as ever, be confronted with lists which state that Great Britain possesses so many battleships, that France, Germany, the United States of America, &c., possess so many; and starting from this hypothesis, articles will be written to prove that either now, or at some date in the comparatively near future, the first-named Power will be overtaken and passed, and that the sea supremacy, which none deny to her in the present, will then be dangerously jeopardized. But the writers and speakers who merely count

units—units which are described as battleships—do not always analyze, and lay before the public which they address, the data concerning the vessels mentioned in these lists.

If a comparison is to be of any value, it must take into account not only the number of ships, but also their fighting capacity. As an example let us see in what way the Channel Fleet, under the command of Lord Charles Beresford, will be constituted in the immediate future. It will consist of eight ships of the *King Edward VII* type and six of the *Formidable* type, fourteen battleships, forming the most powerful and also the most homogeneous force, which is an all-important matter, in the whole world. Of the first type mentioned, the oldest was launched in 1903, the youngest—the *Hibernia*—in 1905. These eight ships each carry four 12-inch, four 9.2-inch and ten 6-inch guns, as well as small guns and five submerged torpedo tubes; their armor is 9-inch Krupp amidships, tapering to 6-inch and 2-inch; their horsepower is 18,000, and their speed nearly 19 knots, their tonnage being 16,350 tons. The *Formidables* are older, November, 1898, being the date of the oldest of the class, and their size is smaller, being 15,000 tons, carrying four 12-inch, twelve 6-inch, besides small guns and four submerged tubes; the horsepower is 15,000, and speed 18 knots. These details are given in order that comparisons may be instituted between this, the principal British Fleet, and the ships of foreign Powers; showing the relative strength of the units which go to make up their fleets, as well as our own. The Home Fleet, which contains the *Dreadnought,* and the Atlantic Fleet are purposely left on one side in order to avoid confusion.

The Navy of the United States of America has, of late years, outstripped that of France and taken second place, next to our own. Undoubtedly, however, the *Dreadnought* took the Americans by surprise, and since her trials and the demonstration that this new departure was a success, they have been attempting to make up their leeway. If we go back to 1898, the date of the oldest of the *Formidables,* we shall find that the United States built in that year the *Kearsarge* and *Kentucky,*

of 11,500 tons, carrying four 13-inch, four 8-inch and fourteen 6-inch guns, with a 16½-inch Harvey-nickel armor belt amidships; the horsepower is 10,500, and speed 16 knots. In the same year were built the practically identical ships *Alabama, Illinois* and *Wisconsin*—as far as tonnage was concerned—but the armament of which differs considerably from that of the *Kearsarge* and *Kentucky*, as, instead of four 8-inch, they carry fourteen 6-inch guns. It will be observed that the *Formidables* are superior in nearly every respect, and have the enormous advantage of homogeneity. In 1901 the United States built the *Maine* class, of three ships, of a tonnage of 12,500, carrying as their main armament four 12-inch and sixteen 6-inch guns; horsepower, 16,000, and speed, 18 knots. In 1904 came the *New Jersey* class, of five ships, of 14,948 tons, with four 12-inch, eight 8-inch and twelve 6-inch guns; horsepower, 19,000, and speed, 19 knots. In the same year came the *Louisiana* and *Connecticut,* of 16,000 tons, with practically the same armament; and in 1905 came a reversion to a smaller type—the *Idaho* and *Mississippi,* of 13,000 tons, with eight 7-inch guns, instead of twelve 6-inch. The *Kansas* class, of 16,000 tons, also date from 1905, and carry the same guns, except that they have twelve, instead of eight 7-inch guns, and that with a horsepower of 16,500 their speed is 18 knots. In 1906 the first of the *South Carolina* class was laid down; and in this ship, and her sister, the *Michigan,* comes a foreshadowing of the future, as with a displacement of 16,000 tons their armament is eight 12-inch guns, the speed and horsepower remaining the same as in the case of the *Kansas* class. But the *Dreadnought* had seized upon the American imagination, ever prone to indulge itself, both by sea and land, with the biggest thing going, and two ships, forming the *Delaware* class, are to be laid down during the present year; their displacement is reckoned at 20,000 tons, and their armament will be ten 12-inch guns. In considering the available battleships of the United States we shall see that, supposing them all to be in commission and ready for sea—which they are not—they could oppose 21 to the 14 of our Channel Fleet (which are ready and in commission), that five

of these are vessels of 11,500 tons, and that homogeneity would be far to seek. Of personnel it may be stated in passing that the United States Navy is short some thousands of men; and although the officers responsible say that this deficit could easily be filled, it cannot be considered that the Navy is an efficient service until this is done; and, in spite of official optimism, the supply of free-born white Americans who seek to gain a living on the sea is by no means on the increase.

The importance of the French Navy has been always reckoned as one of the most potent factors in keeping the balance of European armaments; but it cannot be said that of late years the great Republic has kept up her ancient fame in this respect. This is no place to deal with the political aspect of affairs, but the man who has followed with intelligent interest the doings in the French Navy, and the French arsenals, for the last five or six years, would be blind and deaf to patent facts did he not recognize the enormous mischief wrought, both in the disciplined service and in the great industrial undertakings with which its upkeep is maintained, at the hands of a socialistic Minister of Marine. Dearly is M. Gaston Thomson paying for the policy of his predecessor M. Camille Pelletan, the Socialist, and dearly are the country and the Government paying in bad discipline and slackness among its employés, as well as in the more vulgar and tangible asset of hard cash. Thus it is that in relation to the French fleet of the present day one finds some difficulty when it comes to comparison of battleship strength, as no battleships were laid down for more than three years, and the last of the 1900 program has only just left Saint-Nazaire for Brest, to begin her trials. Not only was incalculable mischief done to discipline and the morale of the fleet and dockyards—witness the strike of the Inscrits Maritimes at Toulon three summers ago, and the antics of that most militant union, the "Syndicat Rouge," in the dockyards; but also the Minister considered that, in addition to anything else, it behooved him personally to advise as to the class of vessel upon which the naval credits voted were to be spent. M. Pelletan decided against battle-

ships, and perpetuated the error of Gabriel Charmes and the "jeune marine" of the early 'eighties, who declared that the day of the "capital ship" had come to an end as soon as the automobile torpedo became a practical and accomplished fact. Gabriel Charmes and his school pinned their faith upon torpedo boats of great speed, but the sea soon demonstrated that such craft were of small value in heavy weather. The submarine, in like manner, became the obsession of M. Pelletan, and on the submarine and the "petite marine," or small craft, has all the money been spent of late years. At last, however, saner counsels have prevailed, and France has started to make up all the leeway that she has lost.

It is impossible in the present day to consider that the *Charlemagne* class, of three ships which date from 1895, are of very much fighting value. They are, however, still on the active strength of the Navy; they are 11,200 tons, and carry four 12-inch, and ten 5.5-inch guns. The *Iéna,* dating 1898, was, as all the world knows, blown up at Toulon in March of the present year. The *Suffren,* 1899, is 12,750 tons, and carries four 12-inch and ten 6.4-inch guns; horsepower, 16,200; speed, 18 knots. The list closes with the *Republique* (1902) and *Patrie* (1903), of 14,865 tons, carrying four 12-inch and eighteen 6.4-inch guns; and the *Liberté* class, of four ships, of 14,900 tons, with four 12-inch and ten 7.6-inch guns; horsepower, 18,000; speed, 18 knots. But "mastodonte," as the French aptly call the craze for big ships, has seized also upon the Republic. At last common sense has regained her sway, and with it recognition of the fact that to be strong on the sea you must possess ships capable of lying in the line-of-battle. Therefore there have been laid down six ships—the *Danton* class, of 18,400 tons, to carry four 12-inch and twelve 9.4-inch guns. Further than this, it has been recognized that the worst of all economies is to keep ships too long upon the stocks, and in consequence, instead of seven years, it is designed that the *Danton* and her five sister ships shall pass into the active service four years from the date upon which their keel plates were laid.

Their fleet, both war and mercantile, is among those things

of which the Germans are the most proud—and with reason. The Kaiser on a memorable occasion declared that "the future of Germany is on the sea," and to do him justice, he has never ceased, in season and out of season, to do his best to ensure that that future shall be prosperous. Germany, like every other nation, piles ship upon ship and gun upon gun, protesting loudly all the while that nothing is further from her thoughts than war; this, of course, is all part of the game, and nobody minds, because we all do and say the same thing. But it may be permissible to doubt whether Germany does not view the development of "mastodonte" in warships with greater dismay than any other nation. The Fatherland was really getting along very nicely in the naval way until the arrival of the *Dreadnought* put all her plans astray. There is one thing that neither King, Kaiser nor Republican President can alter, and that is the physical configuration of the land in which they dwell, and Nature has ensconced Germany behind one of the most intricate and tortuous labyrinths of sandbanks which exists in the world. Very useful are such natural defences against a potential maritime enemy, but when a nation wishes to develop into a great maritime Power, they embarrass it almost as much as they would do its foe. As at present constituted, the Germany Navy possesses no battleship of over 13,200 tons, and has afloat no gun of a greater caliber than 11-inch. There are five of the *Braunschweig* class, dating 1902-3, and five of the *Deutschland* class, dating 1904-6, of this tonnage, carrying four 11-inch and fourteen 6.7-inch guns; horsepower, 16,000; speed, 18 knots. There are also five *Wittelsbachs,* of 11,830 tons, and five of the *Kaiser* class, of 11,150 tons, carrying four 9.4-inch and eighteen 6-inch guns, 15,000 and 14,000 horsepower, and speed 18 knots. The *Brandenberg* class, of four ships, date from 1891, and are of 10,060 tons,

If we compare these ships, of which the German high sea fleet is composed, with the Channel Fleet alone, we need not, at all events, fear the comparison. But Germany has further ambitions, and the "mastodons" which she has projected are

the *Sachsen, Baiern, Baden* and *Wurtemburg.* They are to have a tonnage, it is reported, of 17,710 tons, possibly 19,000 tons—but apparently nothing definite has been settled—and are to carry sixteen 11-inch guns.

The most modern battleships possessed by Japan are the *Kashima* and *Katori,* dating 1905; they are 16,400 tons, carry four 12-inch, four 10-inch and twelve 6-inch guns, have a 9-inch Krupp belt amidships, horsepower 17,000, and speed 18.5 knots. Our allies, however, are perhaps, further advanced than any other nation except ourselves along the road of big-ship building, as the *Satsuma,* of 18,800 tons, is already in the water, and her sister ship is advancing towards completion. There are also one ship, as yet unnamed, building, and one projected of 20,750 tons.

Of Russia there is but small need to speak at present, as her Navy is in an embryonic stage. Even when it is completed, it will not rank very high in the estimation of the world unless its methods of training are vastly different from those which obtained before the Russo-Jananese war.

Italy projects to lay down three or four ships of 16,000 tons; but those remarkably able men, the Italian constructors, have always held that for their country moderate tonnage would suffice, and today 16,000 tons represent moderation.

Austria is more modest still, as 14,500 tons is the limit, as far as is known, to which she will go at present in big-ship construction. There are other countries also which possess warships; but, as far as battleships are concerned, they may at present be left on one side.

The conclusion of the whole matter would appear to be that, for the present, England is in a satisfactory position. But the pace is being forced all along the line, and our rivals are striving to overtake and to surpass us. If this fact is kept in mind by our politicians, we need not fear for our supremacy, for we can build faster than any other nation, and certainly quite as well. The danger is that we may presume on our abilities, and defer our preparations too long. Fortunately, all parties are agreed that the British Navy must always remain supreme on the seas.—"Engineering."

POWER ESTIMATING FOR TURBINE STEAMERS.

The fact that the marine turbine cannot be "indicated" in the ordinary sense, and that nearly three-quarters of a million turbine "equivalent horsepower" have been and are being built for merchant vessels alone, makes us ask what is going to become of the Admiralty coefficient? It is unlikely to die out for many years, but for long enough it has been notoriously untrustworthy, especially for fast vessels, and now that "indicated" horsepower bids fair to become a thing of the past for every steamer of 19 knots and upwards, its use may become less frequent, and will certainly entail more caution than ever. Or will it gradually commence a fresh spell of useful existence in a more modern form?

There is one great feature about the adoption of marine turbines, which is that they have made naval architects and marine engineers think on new lines. Empirical methods of design, and a discreet copying of the last similar job, sufficed for the vast majority of work done for the mercantile marine when the comparatively slow-running engine and propeller were adopted. In their case the margins were large in all directions; the propeller was generally nowhere near its breaking-down point, the link motion enabled a greater mean pressure to be used if the required power was more than the estimate, analyses of component losses were almost invariably ignored, and with one or two exceptions, shipbuilders were content to trust the indicator, and base everything thereon. Consequently vital variables remained undisturbed for want of investigation that the turbine has since enforced, and lest anyone should be unduly assisted by the work of his predecessors, providence seems to have carefully arranged that this casual satisfaction with indicated horsepower and speed should also extend to the stokehold, and that the coal consumption should be measured while the water evaporated—and hence boiler efficiency as well—should be left undetermined. Consequently there arose a term—pounds of coal per indicated horsepower hour—that, while based on good experience with innumerable vessels, was absolutely useless for scientific comparison of results. For the

moment we are not so much concerned with the boiler end of estimating as with the term indicated horsepower. For up-to-date fast steamers it is as useless as nominal horsepower, as Kirk's analysis or Rankin's augmented surface methods, excellent as they may have been in their day. Times change, and what we have to face now is the problem of foretelling the effective thrust needed to propel a hull of given dimensions at a given speed. The term that marine engineers will not only have to get into the way of using, but also of estimating, will be effective horsepower, and *not* indicated horsepower. Now, the horsepower actually required to propel a given hull has to overcome the (1) frictional resistance of the hull; (2) the wave-making resistance; (3) the eddy resistance. The first of these can be calculated with considerable accuracy, though it is customary, by taking a higher value for the coefficient of friction than that originally found by Froude, to include eddy resistance with it, owing to the impossibility of separately determining the resistance due to this cause. Wave-making resistance is hard to determine without a tank, though some empirical coefficients do exist that give a close approximation for certain types of vessel, and careful analytical estimates from similar ships help us to determine the percentage of the total that it represents. In the case of tanks, when the resistance of the naked model has been determined, the effective horsepower is first calculated and the probable indicated horsepower arrived at by means of propulsive coefficients made up as follows: About 5 per cent. is added to the E.H.P. of the naked hull for resistance due to appendages—rudder, bossing, shaft brackets, &c.—and the figure thus found is multiplied by about $\frac{1}{.85}$ for engine efficiency, and about $\frac{1}{.65}$ for propeller efficiency, and by a very small corrective for hull efficiency, that is, for the difference between wake gain and thrust augmentation due to the action of the screw. The net result is that the ratio of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ is about 50 per cent., but varies very con-

siderably. For most vessels the value is between 48 per cent. and 56 per cent. at full speed, but instances of greater and lesser values could easily be quoted. It is an interesting question as to how accurate tank horsepower is. Models of the same ship tried in different tanks give amazingly variable results, but with care and experience the resistance can be determined very closely.

Consider for the moment the analysis of power from the engine end. The indicated horsepower as given by the indicator is considerably in excess of that delivered to the propeller boss on account of friction in the moving parts. Until the genesis of the torsion meter about four years ago it was very difficult to say what this loss was, as the testing of large marine engines against a brake was prohibitively costly. It was generally assumed—largely on the basis of trials of land engines against brakes or generators—to vary from 10 to 16 per cent., and it came as a considerable surprise to many when the Vulcan Company proved the mechanical efficiency of the main engines of the *Kaiser Wilhelm II* to be as high as 94 per cent. in service. Even assuming the accuracy of tank trials, it was therefore impossible to determine propeller efficiency correctly owing to the uncertainty as to the power actually delivered to the screw. Now that this can be done with comparative accuracy, it is so far only attempted on turbine steamers, and we find ourselves in the position of gradually commencing to accumulate isolated facts on the subject, but without any means of comparing them with reciprocating practice, as torsion meters are very rarely applied to piston engines. When, therefore, it becomes necessary to estimate the power required for a turbine steamer, and consequently revolutions of shafts, weight of turbines, &c., there is no guide established yet by past practice enabling us to decide exactly what brake horsepower is required. If a tank is available, or rather if a store of tank data is at hand to estimate from, the matter is simplified at once. If not, the procedure is guesswork. An Admiralty coefficient for indicated horsepower is taken as if piston engines were to be used, and an equally approximate propulsive coefficient is

assumed in order to arrive at the E.H.P. Consequently we find some cases of large margins, and others cut very fine indeed.

With the improvement in steamship performance the value of the coefficient should be rising in ships of all types; but as far as Channel steamers are concerned, there is now no indicated horsepower, and hence no trustworthy coefficient. The solution seems to be to substitute E.H.P. for I.H.P. wherever it can be done. It applies equally to both types of engine, and obviates the inclusion of undetermined losses in important estimates. The known propulsive efficiencies can be used, but the absolute values will change. Instead of taking, say, 240 as a suitable Channel steamer coefficient whose propulsive efficiency is known to be 54 per cent., our value in future will be 444. Unless some step like this is taken, and a new series of values compiled for a most useful formula, we are afraid that for all fast steamers its use and value for power estimating will rapidly wane. The same arguments apply equally well to proportions of boilers, to fuel consumption or relative weights; but at the present time we are in a transitory state on this question, and it is by no means easy to change over from the old-established symbols.—"The Engineer."

THE FRENCH NAVAL ARMAMENT.

The program of naval construction for 1909 now under consideration by the Conseil Supérieur de la Marine promises to be an unusually important one. In view of the limited expenditure upon new vessels in the coming year, it was anticipated that the Minister of the Marine, by restricting his efforts to the six battleships of 18,350 tons now being built under last year's program, was preparing for a much bigger development in the early future. The necessity for clearing off arrears and carrying out reforms in the arsenals and shipyards appeared sufficient to justify the small number of units which is to be put on the stocks during 1908, but a still more important factor

in determining the Minister to postpone the final preparation of his program is the uncertainty that has existed concerning the armament that should be carried on future battleships. While submarines, submersibles and torpedo boats will continue to form an important part of the French Navy, it is obvious that so far as heavy units are concerned the Conseil Supérieur de la Marine intends to follow the lead of other nations by equipping the fleet with as many powerful battleships as possible. It is doubtful whether many, if any, armored cruisers will be included in the list of constructions for 1909, which seems likely to be almost entirely comprised of battleships of 20,000 to 21,000 tons. Fitted with turbine machinery, they are expected to have a speed of 20 knots. Before deciding upon the dimensions to be given to these new heavy units, the Conseil Supérieur de la Marine is now holding frequent sittings with a view to discussing the armament that will be carried. It appears that the Conseil has five proposals under consideration. The first is to equip the future battleship with 12 guns of 305 mm. bore; the second, to increase the number of these guns to 14; the third, to employ 16 guns of 274 mm. bore; the fourth, to have 20 guns of 240 mm. bore; and the fifth, to adopt a compromise with 8 guns of 305 mm. and 8 of 240 mm. bore. There is apparently a strong feeling in favor of the third of these proposals—that is to say, adopting a numerous armament of 274 mm. guns—because it is argued that this type of ordnance is capable of proving very effective at a distance of 7,000 m., which is regarded as the extreme fighting range. It is claimed that at a distance of 7,000 m. the gun of 274 mm. bore will pierce armor 330 mm. thick under an incidence of 20 degrees. The advantage of having sixteen guns of the same caliber is that a greater quantity of shells can be carried than would be the case if there were 305 mm. guns on board, while the number would permit of a concentrated firing which might prove very deadly at the maximum fighting range. All this depends, however, upon whether the future developments in the protection of battleships will not nullify the effect of shells from 274 mm. guns at the extreme fighting

range. As it is expected that the Conseil Supérieur de la Marine will very shortly come to a final decision upon this question of armament, it will be interesting to see whether it has more faith in the guns or in defensive armor. Whatever solution is come to, it is now an article of faith in naval artillery circles that the gun caliber on battleships should be as uniform as possible, whether there is a tendency to suppress guns of medium caliber and employ little or nothing between the heavy ordnance and the light quick-firing guns for resisting torpedo attacks.—"The Engineer."

NAVAL WORK ON THE CLYDE.

With the consignment to the water of the first-class cruiser *Inflexible* from the stocks of John Brown & Co., Limited, Clydebank, on June 26th, naval work on the Clyde, so far at least as the building berths are concerned, became a negative quantity. Little has happened since to change this condition of affairs; but a number of smaller miscellaneous commissions from Admiralty quarters have kept whetting the shipbuilders' appetite for "more." At the moment of writing also a statement is published, on what seems trustworthy authority, to the effect that a well-known shipbuilding and engineering firm on the upper reaches of the river whose building stocks are almost entirely denuded of work of any kind has just booked, amongst other important orders for merchant steamships, a contract to build and engine for the Japanese Government a battleship of about 18,000 tons displacement and of great beam. This requires stronger confirmation, but everything points to its being true in essentials, though details which are not here repeated may be wanting in accuracy.

The *Inflexible,* owing mainly to the discovery of some flaws in her stern-tube castings shortly before the time appointed for launching, and to the delay which the removal of the imperfect and the fitting of perfect castings caused, was some three months later in being sent off the stocks than her sister ship *Indomitable,* which the Fairfield Company

launched in March. The work on other parts of the Clydebank vessel, however, went on as usual, with the result that the cruiser when launched in June, almost three months after the time originally fixed, was in a correspondingly advanced state. Boilers and machinery have been put on board with despatch, and it is hoped that she will be ready for steam trials in six or seven months' time. In all probability the three sister ships—the *Indomitable,* whose fitting out at Fairfield basin is proceeding apace, and the *Invincible* on the Tyne—will all be put into commission during August, 1908.

The battleship *Agamemnon,* which her builders, William Beardmore & Co., Limited, Dalmuir, took down the Clyde about the end of July, after drydocking at Govan for hull cleaning, at the Tail of the Bank, took in coal, adjusted compasses, &c. On August 5th she was boarded by a navigation crew sent from Devonport, and on the 19th she went on a preliminary trial on the Firth. Next day she proceeded down Channel on a thirty-hours' trial at one-fifth power, and after returning to the anchorage it is intended that she shall proceed on a further thirty-hours' trial at seven-tenths power.

A small but interesting proportion of the new work recently placed with Clyde builders consists of Admiralty contracts. The Ardrossan Dry Dock and Shipbuilding Company, Limited, has secured an order from the Admiralty for three horse boats, and the Grangemouth and Greenock Dockyard Company has been entrusted by the Admiralty with the order for five vessels for transport purposes. The latter order will be executed by the company at its Grangemouth yard, and the vessels will be towed to Portsmouth. Some time ago this company contracted to build for the dockyard at Malta three caissons, and to equip them complete with hydraulic machinery. These have now been completed and duly tested in the presence of Admiralty and Dockyard officials. Tested under hydraulic pressure corresponding to the duty they will have to undergo when in position, each of the caissons was opened and closed in the space of 4½ minutes, or half a minute under con-

tract time. The dimensions of the caissons are 96 feet by 42 feet by 22 feet each, weighing 750 tons. They have lowering and lifting bridges on the "Kinniple" principle.

Although not yet in full swing, the new works established on the Clyde, at Scotstoun, by Yarrow & Co., Limited, builders of torpedo boats and other light and high-speed craft, may be said to have been set in motion. The keels of two torpedo boats for the Government have been laid, and other berths are being prepared. The shipyard has a river frontage of about 780 feet, and a breadth of about 710 feet, and the firm has the option of purchasing more ground to the east. Seven slips are being laid out, and the general equipment of the works should make the yard one of the most perfect of the kind in the world. The extensive shops of the engineering and boiler-making departments are being rapidly equipped with the most up-to-date machinery. A feature of the works will be the fitting-out basin, which is now nearing completion. It is 300 feet in length, by 80 feet in width, and it is being covered in such a way as to allow fitting-out work to proceed in all kinds of weather. When the boiler-making shops—to deal with the full demand for the Yarrow water-tube boilers—and the several other departments are fully set going, it is understood that fully 2,000 workmen will be employed.—"The Engineer."

ECONOMY TESTS OF A 7,500 KW. WESTINGHOUSE-PARSONS STEAM TURBINE.

By J. R. Bibbins, Jr., M. Am. Soc. M. E.

The data herein presented comprises the principal results obtained on Sept. 1, 1907, during an 8-hour economy test upon turbine No. 253, installed earlier in the year, at Waterside station No. 2, of the New York Edison Co., New York City. This test was conducted entirely by the New York Edison Co., under the direction of Mr. J. P. Sparrow, Chief Engineer. The various arrangements therefor were carried out in accordance with a mutual agreement between builder and operator,

entered into previous to the test, and the results were obtained by independent computation.

The turbine unit tested is a standard Westinghouse type having a maximum rated capacity of 11,250 kw., and was built to operate on 175 pounds steam pressure, 28 inches vacuum and 100 degrees superheat. Under these conditions the turbine unit was guaranteed to have a minimum steam consumption of 15.9 pounds per kw. hour at the generator terminals with a normal speed of 750 r.p.m. Incidentally, the electrical efficiency of the generator was guaranteed to be 97.8 per cent., exclusive of friction and windage, at a load corresponding to that sustained during the test. The results of the test detailed below show an economy about 7.5 per cent. better than the guarantee.

METHODS OF CONDUCTING THE TEST.

During the test period, No. 2 Waterside station sustained practically all of the 25-cycle load on the system. Of this the unit under test carried practically 70 per cent., the other turbine units in the station assuming the remainder. This load was maintained as constant as possible by remote control of the turbine governor by the switchboard operator. Between the first and the last hours of the test the maximum variation in load was held within 4 per cent. above and below mean. During the last hour, however, the load decreased somewhat. Previous to the test this turbine unit had been running on a load of 7,000 kw., which was increased to its test load ten minutes before the start.

Three-phase electrical load was measured by the two-wattmeter method, using two Weston indicating watt-meters of the standard laboratory type. These instruments were calibrated at the New York Electrical Testing Laboratories immediately before and after the test. Power factor was maintained substantially at unity, and all electrical readings were taken at one-minute intervals.

As a surface condenser was used in connection with this turbine unit, the water rate was determined by weighing the condensed steam delivered from the condenser hot well. This

condensation was weighed in a tank mounted upon platform scales, with a reservoir above large enough to hold the condensation accumulating between each weighing.

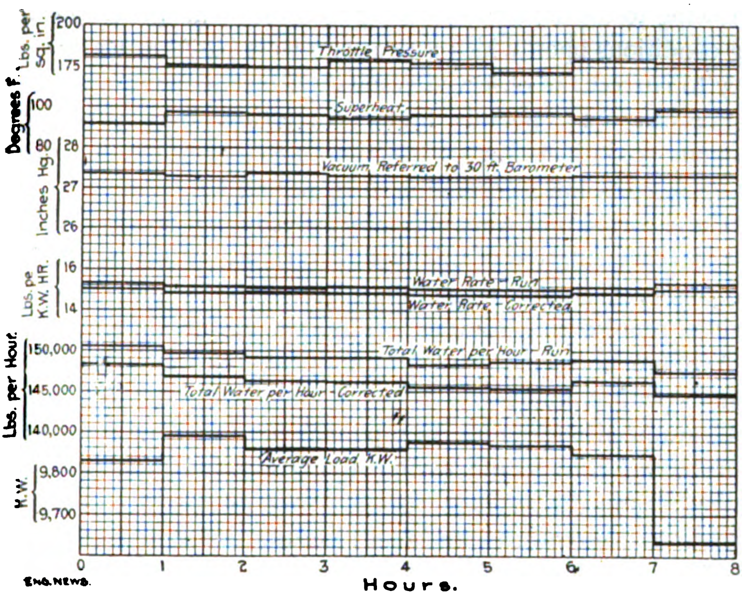
These weighings of 12,000 to 13,000 pounds each were made at intervals of 5 minutes. By a loop method of connecting the gland water supply, the necessity for correcting condensation by an amount equivalent to the weight of the gland water used is avoided.

As the circulating water is quite salt, any condenser leakage may immediately be detected by the salinity of the condensed steam, which should be pure distilled water. On this account condenser leakage was determined entirely by chemical analysis, employing the silver-nitrate test with a suitable color indicator. This method proved extremely sensitive and possessed a decided advantage over the ordinary method of weighing the leakage accumulating during a definite period when the condenser is idle and under full vacuum. As samples of circulating water and condensed steam could be taken at the same time, this method made it possible to discover any change in the rate of condenser leakage taking place during the test, while the method of weighing above described provides only an average result during the period.

In this condensing plant the delivery of the hot-well pump is automatically controlled by a float valve in the interior of the hot well. This maintains the water level therein at a practically constant point, and hence no correction had to be made for difference in level of water in the hot well before and after the test.

Steam pressures and temperatures were determined close to the turbine throttle. As usual, the degree of superheat was obtained by subtracting from the actual steam temperature the temperature of saturated steam at the corresponding pressure carried at the time. All gauges and thermometers were calibrated previous to the test at the U. S. Testing Bureau. It will be noted that both pressure and superheat were somewhat below the guarantee.

Vacuum was measured directly at the turbine exhaust by



PLOTTED LOG OF TESTS OF A 7,500-KW. WESTINGHOUSE-PARSONS STEAM TURBINE.

means of a mercury column with a barometer alongside for reducing to standard barometer—30 inches. This also obviated the necessity for temperature correction between the two mercury columns. During the test the vacuum was not maintained quite up to normal.

RESULTS OF TESTS.

The following data represents the results of the tests, calculated for the conditions as actually run, i. e., for instrumental errors only:

Duration of test, 9:30 A. M. to 5:30 P. M.

Average steam pressure at throttle, pounds, per square inch gauge, 177.5.

Average superheat at throttle, degrees F., 95.74.

Average vacuum (referred to 30 inch barom.) in Hg., 27.31.

Average load on generator, kw., 9,830.48.

Average steam consumption, as tested, pounds, per kw.-hour, 15.15.

Owing to the departure, during the test, from specific operating conditions upon which guarantees were based, it was necessary to correct the observed results by the following amounts:

Pressure (2.5 pounds high) correction, 0.25 per cent.; vacuum (0.69-inch low) correction, 1.84 per cent.; superheat (4.26 degrees low) correction, 0.29 per cent.

These corrections were mutually agreed upon previous to the test as representative of this type of turbine. When applied to the observed steam consumption given above, the following results, representing contract conditions, are obtained:

Average corrected water rate during 8-hour test, 14.85 pounds per kw.-hour.

Guaranteed water rate, 15.9 pounds per kw.-hour.

Referring now to the accompanying log, it is interesting as a check upon the average figures above presented to observe the results segregated into hourly periods, as shown. Here it will be noted that the load was considerably lower during the first and last hour than during the main part of the test. Neglecting, therefore, these two hours and considering only the six-hour period from 10:30 A. M. to 4:30 P. M., the results are as follows:

Average corrected water rate, 14.8 pounds per kw.-hour.

Equivalent water rate, 10.65 pounds per B.HP.-hour.

Equivalent water rate, 9.8 pounds per I.HP.-hour.

The two latter quantities are determined by applying conversion factors for generator efficiency and for internal losses.

In connection with these tests a noteworthy agreement exists between the results noted and those previously obtained from tests of machines of similar design installed in the Manhattan station of the Interborough Rapid Transit Co., New York, and the Long Island City station of the Pennsylvania R. R. At the same loads and with equivalent operating conditions the performance of the machines is almost identical. These economic results, while not exceeding in actual steam consumption to best records of European practice, yet are considered extremely good by the manufacturers, in view of the operating

conditions under which the test was conducted and are held to represent the best results that have yet been obtained by any turbine under the conditions named.

THE FUEL-TESTING PLANT OF THE UNITED STATES GEOLOGICAL SURVEY AT NORFOLK, VA.

By HERBERT M. WILSON, M. Am. Soc. C. E., Chief Engineer of Technologic Branch, U. S. Geological Survey, Washington, D. C.

In April last the Secretary of the Interior organized a new branch of the United States Geological Survey known as the Technologic Branch. Into its charge has been put the work of fuel testing, which was commenced at the St. Louis Exposition in 1904, and the work of structural material testing, which has been in progress at the Forest Park plant at St. Louis since 1905. In addition to the straight fuel testing, investigations of the occurrence of fuels—chiefly on the public lands of the West—are being carried out; investigations into better methods of firing coal, and into the variety of furnace in which certain coals should be utilized, with a view to the abatement of smoke and best utilization of heat; records and investigations into waste of coal due to mining explosions; the causes of the latter and their possible prevention; the better use of explosives and the use of better explosives in coal mining; investigations into the mining engineering features of coal waste, due to bad methods of mining, handling and marketing the product; investigations into the occurrence, preparation and utilization of lignites and of peat as a fuel; and tests of the relative efficiency and commercial value of liquid and mineral fuels as utilized in internal-combustion engines—chiefly gasoline, kerosene and denatured alcohol.

During the past three years the testing of the fuels of the Middle West has been carried on at the St. Louis plant, and a complete survey of the fuels of the United States in the matter of chemical composition and combustion tests has been made in a preliminary way.

In all something less than 200 separate fuel deposits from Pennsylvania to California have been so tested. At the time of the organization of the Technologic Branch it was decided to remove the fuel-testing plant from St. Louis to Norfolk, Va., with a view to making more extended tests on the fuels reaching tidewater for shipment to the naval and merchant marine of the world. These tests are planned especially with a view to indicating the general methods of utilization of the fuels from the New River, George's Creek and Pocahontas coal fields of Maryland, Virginia and West Virginia, in the hope of securing increased efficiency and of preventing waste and of thus extending the life of these coal deposits. This work has especially in view the needs of the Navy of the United States.

The Norfolk plant is located within the grounds of the Jamestown Exposition. This location was selected because of convenience in the way of transportation facilities furnished by the Tidewater Railway, now known as the Virginian Railway, and because the Exposition Company erected a building for the occupancy of the fuel-testing plant, thus saving that expenditure to the Government. This building is situated to the right of the main entrance to the Exposition grounds, near the Service Building, and is admirably suited to the work. It is a one-story building, 325 feet in length by 125 feet in extreme width, and 44 feet in clear maximum height. One end, 75 feet in width, separated from the remainder by a brick fire wall, is devoted to the briquetting plant. A portion of the same wing, enclosed in brick fire walls, is occupied as a boiler room. The main part of the building is occupied by gas and gasoline engines, turbo-generators, chemical laboratories, offices, etc. A portion of the remaining wing is occupied by a section of the structural-materials testing laboratory and by offices; and the remainder of this wing by a commercial exhibit of denatured alcohol appliances in space not immediately required by the Federal testing plant.

Tests of fuels are being made on long runs, on one type of fuel at a time, the tests extending over a sufficient period to

permit of their approaching commercial conditions. For example, gas-producer tests made at the St. Louis laboratory were on runs of about 48 to 60 hours, and the maximum commercial conditions were only barely reached in the operation of the producers when the test would have to be discontinued. The results, while exceedingly valuable as indicating the relative efficiency of fuel in the gas producer or under the steam boiler, are being supplemented by the more prolonged tests being made at the Norfolk plant. Each type of fuel, such as that from a mine typical of a given coal seam is tested in lots of about 150 to 250 tons. A portion is converted into producer gas, and as the test progresses the best method of handling and firing to produce the maximum of high-grade gas is ascertained.

This plant consists of two Taylor pressure-gas producers; the inside lining of the brick jackets is 7 feet high, and the height to the top of the casing 15 feet. The gas from the producer passes through a tar extractor and a scrubber, and then through a purifier into the gas holder. In operating this producer during several years on various types of soft coal, slack, lignite and peat, no difficulty has been found in working effectively, regardless of the amount of volatile matter distilled from the fuel. The gas holder has a capacity of 4,000 cubic feet. From it the gas passes through a meter and operates a Westinghouse gas engine of 235 H.P. The power generated is converted into electrical energy by belting to a 200-kw. Bullock generator, and is measured on a switchboard from which the load can be regulated so as to maintain a steady full-load value.

Among the determinations made in this section are: The length of the test run; the effect of the size of coal; the maximum returns from different fuels; the response of the producer plant to sudden changes in demand for power; the effect of rapid load variations; the best depth of fuel bed, etc. In the course of the test, time is kept of the labor force, of the weight of all fuel, of ashes, of the fuel consumed in the auxiliary boiler, the amount and value of the water consumed, and every item which goes to make up the cost of producing 1 H.P. from a given quantity of fuel.

Every shipment of fuel which is tested for producer gas is also tested for combustion in the steam section; and a portion of it is manufactured into briquettes, which are then tested in comparison with the run-of-mine coal from which they are manufactured, under the same boilers and by identical tests.

The steam section contains two Heine boilers of about 210 H.P. each, and one Babcock & Wilcox marine-type water-tube boiler of 250 H.P. One of the three furnaces is equipped with a Roney overfeed stoker, one with a Jones underfeed stoker, and one for hand stoking only. The three may be operated on natural draft, forced draft, or through the Green Fuel Economizer Co.'s induced-draft fan. The baffling in the different furnaces is arranged differently, so as to procure different lengths of travel of the gases of combustion. It is thus possible to test the same fuel in different sizes, with different methods of stoking and draft, and to baffle in such a manner as to determine the most economical performance under different rates of combustion and the best arrangement of grate and heating surfaces.

The steam generated is used in operating two direct-current DeLaval Turbo-generator sets, each of which is rated at 300 H.P., and is of a speed of about 9,000 r.p.m. The electricity generated, like that generated from the producer gas, is measured and controlled at the switchboard. The work of the steam section is so planned that the performance of the boiler is successfully isolated from the combined performance of the boiler and furnaces, and the tests are being continued with the object of determining still further the efficiency and performance of the furnace alone. The tests are also conducted with a view to determining the method of firing various fuels, so as to reduce to a minimum the smoke emitted from the stack. Both in the producers and in the boiler furnaces holes have been bored at various heights to permit of measuring the highest temperature in various portions of the combustion chambers and fuel beds, and for the extraction of samples of gas for analysis.

Included in the plant are a research chemical laboratory and

a research physical laboratory, for conducting investigations into the gases of combustion. In addition to these research laboratories there is a small commercial gas laboratory for analysis of the gas produced, and a small chemical laboratory for analyzing the fuels and ashes in the steam plant.

The fuels consumed at the plant are inspected at the mine before shipment, and samples are sent to the chemical laboratory for analysis. Similar samples are taken from the car lots as unloaded, and from time to time as the fuel is fed to the producers and the boilers. By this means it is possible to secure, by chemical analysis, data which will permit of determining the values of fuels and their probable performance without test, by comparison with the behavior of those which have been tested and analyzed.

The briquetting room is equipped with a large American briquetting machine of the Renfrow type, and with an English briquetting machine of the Johnson type. The briquettes of the former weigh less than a pound each, are of flattened hemispherical section, and are about the size of furnace anthracite. The briquettes made on the English machine are rectangular, about 3 by 8 inches, and weigh several pounds apiece. Both briquetting machines are operated by a 50-H. P. motor, the fuel being received from overhead bins on a mixing platform; there the coal and the binder (usually about 5 to 6 per cent. water-gas pitch) are mixed, and pass through a disintegrator; they are thence elevated into the tops of the briquetting machines, where they come in contact with live steam at a high degree of superheat. Thence the mixture passes into the briquetting machines, and is pressed into briquettes at pressures ranging from 1,000 to 2,000 pounds per square inch.

Experiments with various binding materials have been made, and are being continued in the experimental laboratory, located at the plant, in which both physical and chemical tests and analyses of briquettes are being conducted. At present the briquetting plant is occupied largely in the manufacture of briquettes from Pocahontas coal, for testing upon naval ves-

sels. One hundred and fifty tons of Pocahontas coal have just been briquetted and delivered at the Navy Yard at Norfolk for testing on a battleship, in comparison with a similar amount of anthracite culm briquetted, and with anthracite run-of-mine and Pocahontas run-of-mine. Similar briquettes are to be furnished for testing under Norment water-tube boilers upon a torpedo vessel, and under Scotch boilers upon a naval tug.

It is believed that these fuels when briquetted will produce a little higher efficiency and develop somewhat more power in a given furnace than when not briquetted, and that they will burn practically without smoke. During the past few months extensive experiments on briquetted fuels, made at this plant, have been conducted in cooperation with the Pennsylvania R. R., the Missouri Pacific Railway, and other roads. The results of the tests made under the direction of Mr. A. W. Gibbs, General Superintendent of Motive Power of the Pennsylvania R. R., have been most gratifying in indications of the probable success to be attained in using briquetted coals, of even comparatively low grade, in producing with a given locomotive a higher efficiency than can be produced by run-of-mine coal; and also in the elimination of smoke by their use, under normal conditions of running.

In the liquid-fuel section, work of much importance is in progress in the determination of the relative availability and efficiency of gasoline, kerosene and alcohol. The equipment includes two 15-H.P. Otto gasoline engines and two 15-H.P. Nash gasoline engines; there are also a 2-H.P. and a 15-H.P. International Harvester Company's gasoline engine. Experiments are being conducted on the relative performances of gasoline and of kerosene, and of both as compared with alcohol. The experiments cover the whole range of this field, including the examination of the different carburetors with a view to determining the most efficient method of vaporization. As with the other tests, exact record is kept of the amount of each fuel used, the labor and water, etc., so as to determine the exact cost of producing a unit amount of power.

The fuel-testing plant is equipped with modern coal-handling

apparatus. Hopper dumping cars are switched on to a track adjoining the plant, and the fuel is dumped into a pit, whence it is lifted 40 feet by a link-belt elevator; from this it can be discharged either through a chute into the gas-producer bins, which are near by and on a level with the top of the producers, or into the bins of the boiler plant, or on to a Robins conveying belt which delivers the coal into one of two storage bins located under the roof of the briquetting plant.

Tests of lignites, both for producer gas and for briquetting, will be conducted in this plant, which possesses a large German lignite-briquetting machine. This lignite press has, however, not yet been installed, and it will doubtless be some time before funds will be available to permit its operation.

Tests of peat are being made in connection with a complete survey of the peat deposits of the Atlantic Coats. Several carloads of Dismal Swamp peat will be macerated in the disintegrator on the plant, dried and converted into producer gas.

In addition to the probable lines of tests in connection with the survey of the fuel deposits of the United States, which are being conducted at the Norfolk plant, the Technologic Branch of the Geological Survey has at Denver a plant equipped for tests of the availability of the western coals for coking purposes—especially those on Government public lands. This plant is equipped with a washery provided with different types of jigs and with means for float and sink tests, etc.; the washed coal is thence conveyed to two standard beehive coke ovens. It is believed that the extended series of tests being conducted at this plant will result in improving the quality of the output of coke produced from western coals.

—"Engineering News."

CECILIE—THE FASTEST RECIPROCATING-ENGINE LINER AFLOAT.

The arrival at the port of New York of the transatlantic liner *Kronprinzessin Cecilie,* of the North German Lloyd Steamship Company, marks the advent of the last and finest of that great quartette of high-speed ocean steamers of this

company, which has helped so greatly to advance the speed and comfort of transatlantic travel. Commencing with the *Kaiser Wilhelm,* which was the first ship to maintain an average of over 23 knots an hour across the Atlantic, the company have placed in service at intervals of a year or two the *Kronprinz Wilhelm,* with a record of 23.47 knots, and the *Kaiser Wilhelm II,* which raised the speed to 23.58 knots, the present record of the Atlantic. The last-named ship, which was brought out in 1904, proved to be so eminently satisfactory that, when the company decided to build the *Cecilie,* they considered that they could not do better than duplicate the *Kaiser Wilhelm II* in every particular. This was done; and that the ship will equal, and probably exceed, the performance of the sister vessel is shown by the fact that on the trial trip, over a measured course of 60 miles, the *Cecilie* averaged a speed of 24.02 knots.

In view of the fact that the *Lusitania* has shown such good results on her trial trip, and is likely to capture the Atlantic record, and that the German Lloyd Company are certain, in the future, to make an effort to win back the record, it is probable that the *Cecilie* is the last high-speed transatlantic steamer of very great power that will be built with reciprocating engines. In fact, it may be taken that in this ship the German shipbuilders have carried the development of the reciprocating marine engine up to the high-water mark of its possibilities. Further advance will be either along the lines of the steam turbine, or of turbo-motors with the electric motors direct connected to the propeller shafts; or it may be that the next advance will be marked by the introduction on a large scale of the marine producer-gas engine. That the limits of the reciprocating engine have been reached is shown by the great magnitude attained by many of the engine parts, and notably by the propeller shafts. These last in the *Cecilie* are 25¼ inches in diameter, and upon each devolves the heavy duty of transmitting at times as much as 24,000 horsepower. This is an enormous load to be imposed, day and night for nearly a week upon a single member, and the question of further increase of power is halted by the serious difficulties that would be encountered

in the forging of shafting of the necessary size, elasticity and durability. For this reason alone naval designers are being driven to the use of three and even four shafts; and to such an arrangement the steam turbine lends itself perhaps more readily than the reciprocating engine. In the new record-breaking ship which the North German Lloyd's company are pretty certain to undertake, it will be a question of great interest as to which of the four leading types of turbine will be adopted—whether the English Parsons, the American Curtis, the German Zoelly, or the French Rateau.

If we are right in our conjecture that the *Cecilie* will mark the highest point to which the development of the reciprocating engine will be carried in high-powered steamers, the *Cecilie* and the sister ship will always be notable landmarks in the future annals of the marine engine. They have set the figures for fuel economy at a point which must ever tax the skill of the steam turbine builders to surpass, if indeed they ever attain it. Before giving any details of the motive power, we may mention that the *Cecilie* is 706 feet long, with a beam of 72 feet, a depth of 44 feet 2 inches, and a displacement of 26,000 tons. She has the usual double bottom, and the hull is divided into twenty watertight compartments. All the bulkhead doors may be closed directly from the bridge; and like all modern liners of the first class, she may be considered unsinkable by any of the ordinary agencies of disaster. Of the interior accommodations of the ship it is sufficient to say that they are superior, even to those of the later ships of this line; while the decorative features are marked by the simplicity and refinement of the latest school of marine decorative work.

The engines of the *Cecilie* are four in number, each carried in its own separate watertight compartment. Each engine has indicated about 12,000 horsepower in actual service. They are placed in pairs in tandem, two on each shaft. In the four engines there are altogether sixteen cylinders—four high-pressure, four first intermediate, four second intermediate and four low-pressure cylinders. The high-pressure cylinders,

which are carried in tandem over the first intermediates are 37⅜ inches in diameter; the first intermediates are 49¼ inches; the second intermediates, 74⅞ inches in diameter, and the low-pressure cylinders reach the truly enormous size of 112¼ inches diameter, these being the largest marine cylinders, we believe, ever built. The common stroke is 6 feet. Steam is supplied from nineteen cylindrical boilers, at a pressure of 230 pounds to the square inch, through four main steam pipes, each of which is 17 inches in diameter. To produce this steam 764 tons of coal are burnt every twenty-four hours in 124 furnaces; and, as we have above stated, the resulting 48,000 horsepower of the main engines has proved sufficient to drive the vessel at slightly over 24 knots an hour.

It is an interesting coincidence that the finest, and what will probably be the last, of the reciprocating-engine liners, should be placed in service at about the same time as the first of the new high-speed turbine liners. The performance of these two ships will naturally be watched with the keenest interest; for in spite of the fact that the turbine-driven ship greatly exceeds the other in size and power, the *Cecilie,* herself, with her displacement of nearly 30,000 tons, and her horsepower of nearly 50,000, is sufficiently large and powerful to eliminate, when driving into a head sea, much of the advantage due to the higher momentum of the larger ship. It is to be hoped that the facts as to relative coal consumption will be made available for the technical world. Just now, the steam turbine is being saddled with a reputation for extreme costliness in fuel; and predictions along these lines are being made so freely regarding the new Cunarders, as to suggest that perhaps the wish, in some quarters, may be father to the thought. For ourselves, we think it is likely that, because of the size of the plant and the high speed at which the turbines are being run. the new Cunarders will show about the same economy as the latest reciprocating-engine ships. This has been brought down in the *Cecilie* to 1.4 pounds of coal per horsepower per hour, including the auxiliaries.—"Scientific American."

THE TURBINE, AND GERMAN LINES.

The German lines are by no means convinced that the turbine is the best form of steamer for oversea purposes. Director Wiegand of the North German Lloyd says that the *Lusitania's* victory is not to be attributed to the fact that she has turbines but that she has such high horsepower—70,000.

The director continues: "The *Lusitania's* turbines, with their relatively small number of revolutions, should not run more economically, to say the least, than a good triple or quadruple-expansion steam engine. The economic value of the turbine lies chiefly in its high revolutions and the satisfactory utilization of steam thus secured. The rapid revolution of the propellers, however, means a reduction of their efficiency. In order to make the propellers revolve at the speed which renders them most effective, it is necessary to reduce the speed of the turbine to a point where it no longer operates economically. After many years' experience it has been found that the propellers of the North-German Lloyd's express steamers work most economically at about 80 revolutions per minute, whereas the *Lusitania's* screws have 180 revolutions. For this reason I do not believe that the *Lusitania's* machinery plant represents an advantage over the prevalent type. In other words, if I build a ship of the same dimensions and put into it an equal power in the form of reciprocating engines, the effective result will in all probability be the same, if not superior to the turbine-equipped steamer. One disadvantage of the turbine steamer is that it possesses inferior maneuvering capacity, because all the steam at hand cannot be used for backing.

"Another question arises: What practical advantage is it to increase the speed of steamers in the North Atlantic traffic to 24 or 25 knots per hour? I am of the opinion that the advantage is just about equal to zero. With our present speed of 22 to 23 knots we arrive at New York by day, whereas we should arrive at night if our steamers should make 24 or 25 knots. Precisely for this reason it was that we brought our express steamers up to 22 and 23 knots. It is well known that we refused to accept the *Kaiser Friedrich* several years ago be-

cause it did not make contract time and would land in New York at night. To bring us any practical advantage in this respect, therefore, the speed of a steamer must not stop with 24 or 25 knots, but would have to make 25½ to 26 knots. That is a circumstance which is not sufficiently taken account of in Germany. We are under obligations to call at Southampton and Cherbourg within certain hours in order to make connections with the trains for London and Paris, and must accordingly stop our steamers at the usual hours, and we could not equalize matters (in case very fast steamers were used) by postponing the sailings for several hours."

Director Wiegand concludes that if the Lloyd should increase the speed of its steamers to 24½ knots their operation would be less economical, since the vessels themselves would cost more, consume more coal, and besides all this would arrive in New York at night, after which they would be compelled to lie at their docks till the regular time for sailing, thus counterbalancing any advantage from greater speed. "Of course," he adds, "if we increase their speed to above 25 knots they would arrive at comfortable hours, but the cost of such a steamer of about $7,500,000, along with the vastly increased expense of operation, would be out of all proportion, economically, to the saving of time at sea. Such a saving would not present a strong enough attraction for the general public to make them willing to pay the increased cost of tickets. Hence, as our company is dependent upon its own resources (instead of subsidies) it could not get any business advantage from putting on such fast steamers."

The report that the vibration on the *Lusitania* is considerably greater than on the fast German steamers has been looked upon here as an additional reason for the German lines to adhere to the old type of engine. Owing in part to that increased vibration the Germans think that this steamer and the *Mauretania* will not prove dangerous competitors for the first-class passenger traffic between Europe and New York. Only when they make much faster time than the *Lusitania* did on her maiden voyage could they become dangerous rivals; but then

the vibration would increase still more annoyingly, so as to neutralize the attraction of quick time; for comfort is, after all, the chief thing desired by the ocean traveler, and a day more or less at sea cuts a very slight figure with everybody except a few hurried business men.

VENTILATION AND REFRIGERATION OF AMMUNITION HOLDS.

Translation of abstract of paper read at the Bordeaux International Congress in Naval Architecture by ADRIEN BOCHET.

The safety of ammunition holds has always engaged the attention of naval architects, and for many years past measures have been taken for isolating these holds, for protecting them against the various causes which may lead to an accident, and for inundating them in the event of danger. Modern powders, and the considerable development in the use of machinery on board ship, have increased the risks in a large proportion. Modern powders have excellent ballistic properties, but are also most unstable, and their instability increases very rapidly with an increase in temperature. By their gradual alteration they also set free a quantity of inflammable gases which may give rise to explosive mixtures. The development in engines and boilers, in auxiliary engines, and in the extent of steam pipes laid throughout the ship, has led to a greater heating of the various compartments. The distribution of the armament, and the necessity of providing ammunition holds in proximity to the guns to be served, have often resulted in the holds being in locations that are particularly unsuitable from the point of view of temperature. Therefore, concurrently with a gradual increase in sensibility to heat, which forms one of the characteristics of modern powders, causes tending to augment the temperature of the ammunition have increased in a large proportion also, and numerous attempts have been made with a view to ensure the artificial refrigeration of the holds.

Cooling by ventilation alone is a simple and safe means to prevent the accumulation of explosive gases, but is quite

insufficient to ensure a decrease in the temperature of the holds when the temperature of the air outside reaches 20 degrees (say 70 degrees F.), and when the causes making for an increase of temperature in the holds attain a certain importance. The heating of the air which enters the ship occurs rapidly on contact with the warm bulkheads, by reason of its low specific heat. A rise of 10 degrees C. has often been noted for courses which appeared as short and direct as possible between the upper works of the ship and the fans placed in the holds. Cooling by the means resorted to in refrigeration holds, such as those for the transport of meat, appeared at first to constitute a final solution of the problem. But this failed completely, owing to the very great difference existing between these holds and those for storing ammunition. In the former there is no ventilation, and they remain closed during the whole passage. They are also maintained at a very low temperature, and one result of this is that the small proportion of moisture contained in the local air on starting gets deposited on the products contained in the refrigerating chamber, and does not injure them in any way. Ammunition holds, on the other hand, require ventilating. Even were one so imprudent as to do away with ventilation completely, or to reduce it in too large a measure, the requirements of the service would demand the frequent opening of the hold, thus allowing every time the outside air to penetrate into it. The temperature which it is suitable to maintain in an ammunition hold is much above 0 degree C. (32 degrees F.). One result of this frequent renewal of the air in an ammunition hold is that the moisture it contains is condensed on all the surfaces which are maintained at a lower degree of temperature than that of the outside air, and the water which cannot be congealed, as in the case of a provision hold, trickles down the partitions of the holds and the ammunition. This very grave disadvantage can only be avoided by cooling the air entering the ammunition hold, so that it is not at a higher temperature than that of the contents or of the walls of the hold.

The only rational means to cool ammunition holds is to ven-

tilate them with air suitably cooled, and this deduction has received the sanction of actual practice, as it is the only method of cooling which has prevailed so far. It has been applied to fourteen French battleships, which carry together forty-three ammunition-hold cooling plants, and to eight Russian battleships, which are fitted with thirty-seven similar plants. It will be fitted also to the armored cruisers *Waldeck-Rousseau* and *Michelet,* now in course of construction. The ammunition holds of these two ships are to be provided with aero refrigerators, in which a circulation of artificially-cooled liquid is maintained constant; the dynamo rooms are to be cooled by a similar installation, but with sea-water circulation.

Until recently, the maximum temperature which was thought advisable for ammunition holds was 35 degrees C. (95 degrees F.); now, however, this limit has been brought down to 30 degrees C. (86 degrees F.). With the former limit direct cooling by sea-water circulation sufficed, as the temperature of the water taken from a certain depth remained perceptibly lower than 35 degrees C. in all parts of the world. The latter limit, however, can only be reached by having recourse to artificial refrigeration.

The refrigerators for cooling the air consist of metallic surfaces, on one side of which the cooling liquid circulates, the air to be cooled circulating on the opposite side. A pump ensures the circulation of the liquid, and a fan that of the air. The complete apparatus are built by F. Fouche, 38, Rue des Ecluses St. Martin, Paris.

The extent to which refrigeration has to be carried depends upon the amount of heat which enters the hold: it is necessary, in the first place, to reduce to the lowest possible minimum this amount of heat. The afflux of heat in the hold is caused by radiation from the warm sides and by conductivity from the metallic pieces they contain, and which are connected to the heated sections of the ship. The methods of obtaining a thermic insulation of the hold are the following: Radiation from the warm sides can be reduced by an inside lining made of substances the conductive and emissive properties of which

are low, such as cork and asbestos. Another solution is to build a double wall, to obtain an air lining, in which case it is advantageous to ensure in the double wall a circulation of air at the lowest possible temperature. The circulation of a cold liquid inside the double wall will give most satisfactory results, provided the liquid be brought sufficiently in contact with all the metallic pieces that are liable to carry heat into the hold by conductivity.

It is easy in principle to ensure the thermic insulation of the hold, but in actual practice the application of the various methods is surrounded with great difficulties, owing to the arrangement of the holds, the small space available, and the necessity of preventing every cause of damage to the ammunition. The use of simple insulating inside coverings of a thickness proportional to the heating of the walls and the available space is evidently the most easy solution of the problem. As this method has hitherto proved of sufficient efficiency, it has alone been developed in actual practice. The use of double walls with air circulation has, however, been successfully combined with the insulation of the sides.

The use of a cold-liquid screen is surrounded by the following difficulties: In order to be efficient, there should be no break or interval in the current. It should reach the whole of the metallic pieces which are liable to cause heat in the hold by conductivity. All risks of inundating the hold in case of leakages have to be removed. The cold screen is constituted by a double wall, but it is absolutely impossible to give the space sufficient dimensions to allow a man to enter it in order to paint the plates and keep them in a good state of repair; this proscribing the use of all liquids, such as sea water, which may corrode the plates. The solution consists in causing to circulate inside the double wall a liquid, such as fresh water charged with oil, or milk of lime, with which there is no risk of corrosion. The total quantity of liquid thus circulating has to be as low as possible, so as to prevent in any case the risk of inundation of the hold. If a leak did occur, a small quantity of liquid only would escape without risk of damaging the ammunition. Sim-

ple means suffice to show when an accident of this kind has happened.

The paper was accompanied by drawings showing how a hold could be insulated and the method followed for ventilating it with cool air. The insulation of the hold is completed by the flowing of the exhausted air round the walls. The liquid and the air for ventilation can be cooled simply by sea water or by means of a refrigerating machine, according to the temperature required to be maintained in the hold.

The amount of heat Q^1 penetrating through the walls is proportional to their surface $\int s$, to the difference of temperature between the two sides $T—t$ and to the time.

The number of heat units going through a given partition per unit of surface and time q^1, for one degree difference in temperature, is determined experimentally. Numerous tests have established the figures corresponding to partitions covered with the usual insulating material. The following formula is obtained:

$$Q^1 = \int s \ (T = t), q^1$$

for the unit of time.

Further, the metallic parts, such as partitions, bulbs, standards, projectile supports riveted to a warm wall, cause a quantity of heat to penetrate the hold; this is

$$Q^2 = \int s \frac{q^2}{e} \ (T - t),$$

where s is the section of the metallic part considered; e the distance between the point where the heat maintained at temperature T penetrates, and the point whence this heat is emitted inside the hold, the temperature of which is t; q^2 being the number of heat units passing per unit of time and per unit of section of the piece, considered between two points, with the unit of length between them, and maintained at a temperature difference of one degree.

Neglecting the amount of heat brought in by the personnel,

caused by the handling of the projectiles, also the entrance of warm air, the total quantity of heat penetrating the hold will be

$$Q = Q^1 + Q^2$$

This amount of heat has to be totally carried away by ventilation in order that the hold may be maintained at the required temperature. It is necessary, therefore, to send into the hold a volume of air V capable of absorbing the quantity of heat Q, by being heated from its entrance temperature γ to an outlet temperature θ. The calorific capacity of air being 0.3 heat unit per cubic meter, the following will be the formula:

$$V = \frac{Q}{0.3\,(\theta - \tau)}.$$

It is necessary to ascertain that the volume of air thus determined meets the ventilating conditions of the hold according to the nature of ammunition it contains.

In order to bring the volume of air V from the initial temperature θ measured at the moment it enters the refrigerating machine to the temperature θ measured at the outlet, the amount of heat given up by the air proper, by the water vapor it will contain after cooling, and that given up by the vapor which condenses, have to be deducted.

The specific heat of dry air being 0.3 heat unit (calorie) per cubic meter,

That of vapor being 0.48 heat unit (calorie) per kilogram.

The latent heat of water evaporation being 606.5 heat units (calories) per kilogram,

p the weight of water vapor contained in one cubic meter of air drawn, and

p^1 that remaining in one cubic meter of the air after cooling The formula for the heat to be absorbed will be

$$Q^1 = V[\theta - \theta\,(0.3 + 0.48p^1 + 606.7\,(p - p^1)].$$

In designing the refrigerating machine it is necessary to add to this amount of heat Q^1 that resulting from the heating of the apparatus themselves, and of the pipes in which cold air and liquid circulate.

The dimensions of the apparatus and of its auxiliary parts—the circulation pump and the fan—are deduced from the volume of air V flowing per unit of time, and from that of the cold liquid necessary to carry away the quantity of heat Q^1 increased as aforesaid.

It should be remarked that the power of transmission of heat of an aero-refrigerator from air to water is, so to speak, without limits; it depends solely upon the output. It increases with the speed, so that the dimensions of the apparatus, suitably proportioned, are solely dependent upon the required pressure of the air and the cooling liquid.

Observations made on a number of installations show that the machines supply air easily at a temperature which does not exceed by much more than about 1 degree C. that of the cooling liquid, whatever be the temperature of the air drawn into the machine. The water-vapor contained in the air is completely condensed in the aero-refrigerator in a proportion which corresponds with the fall in temperature. The hold thus remains perfectly dry.

Tests made on board the *Sully*, in the Far East, have shown that with temperatures of 31 degrees C. (87.8 degrees F.) for sea water, and 36 degrees C. (96.8 degrees F.) for the air, the apparatus maintained the hold at a temperature of 32 to 34 degrees C. (89.6 to 93.2 degrees F.).

The results of other tests, and illustrations of the apparatus, were given in the paper.—"The Marine Review."

NEW FOURTEEN-INCH GUNS FOR COAST DEFENSE.

By Charles A. Sidman.

Plans have been perfected for the new 14-inch caliber guns which are to be added to the existing coast-defense system now in place along the coast from Maine to Washington State, and Gen. Crozier, Chief of Ordnance, United States Army, has the Watervliet arsenal engaged in the manufacture of these new pieces.

The 14-inch gun will be something new in ordnance design, and although fully two inches larger in caliber than the standard coast-defense gun of the first grade (a 12-inch caliber), the new gun will be shorter in length, and the outside diameter will be smaller. The powder chamber will be less than the gun now in use, and it will be lighter in weight.

It is proposed to make a weapon that will have a range and striking force at least equal to the present standard 12-inch gun, but which shall greatly exceed the very limited life of that gun. One strong point also in favor of the new gun is the fact that its first cost will be less than that of the 12-inch gun, while the addition to the life of the piece will result in a greater economy.

In speaking of the new gun, Gen. Crozier said:

"There will be a vast difference between the present standard 12-inch gun and the new 14-inch gun. The 12-inch gun fires a projectile of 1,000 pounds weight, with a velocity of 2,550 feet per second, using 366 pounds of powder, and only has a life of between sixty and seventy rounds, before it has to be relined. This necessitates dismantling and shipping back to the factory for that purpose. The new gun, which is much shorter in length than the 12-inch gun, will fire a 1,600-pound projectile, use nearly 100 pounds of powder less than the 12-inch gun, and only gives a muzzle velocity of 2,150 feet per second, while its life will be nearly four times that of the present standard.

"By reason of the lower velocity required and the consequent smaller charge, it is possible to make the 14-inch gun proportionately shorter than the 12-inch gun, and the smaller charge of powder also involves a less diameter of powder chamber, and, therefore, with the same thickness of wall of the chamber in caliber, a less exterior diameter of the gun over the breech. These elements of saving are so considerable that the weight of the 14-inch becomes actually less than that of the 12-inch; and as a lower maximum pressure is needed, it is possible to attain all the strength which will be used without employing the most expensive steel.

"The muzzle energy of the 14-inch projectile will be about 15 per cent. greater than that of the 12-inch; and because of its lower velocity and its greater weight, the retarding influence of the air will be much less upon this projectile, so that the gain of energy will be in greater proportion with each increment of range. Because of its lighter weight and of the cheaper material of construction, the cost of the new gun will be less than that of the 12-inch gun, while the cost of powder will also be less than that for the 12-inch by about $70.

"The cost of the 14-inch armor-piercing shot will be about $100 more than that of the 12-inch shot, so that the total cost of a single round will be about $30 greater. Taking into consideration, however, the rapid deterioration of the 12-inch gun, and adding the cost of relining to that of the ammunition, which would correspond to the number of rounds making relining necessary, the cost per round, including the deterioration, for the 14-inch gun is only about 68 per cent. of that for the 12-inch gun.

"The penetration of the 14-inch projectile through Krupp armor at 10,000 yards is about 11 inches, while that of the 12-inch is about 10½ inches; while the range at which the 14-inch projectile will penetrate 12-inch Krupp armor is about 8,700 yards, as against 8,500 yards for the 12-inch projectile.

"To sum up, then, it appears that in situations requiring the greatest power, a 14-inch gun, with 2,150 feet per second muzzle velocity of projectile, instead of the 12-inch gun with 2,550 feet per second initial velocity, gives us a lighter gun, a cheaper gun, a heavier projectile, greater muzzle energy, a still greater proportion of energy at each distance beyond the muzzle, and a life four times as long."

Regarding the life of the 12-inch and 14-inch guns, it has been considered that, in attempting to run by fortifications guarding the entrance of a harbor, the period that would elapse from the time that the leading vessel of the fleet would come within range until the last vessel would pass beyond the range of the coast guns would be about two hours. It is therefore evident that a new 12-inch gun would not last through such an

engagement; and considering that this gun is capable of firing for a considerable interval at the rate of forty-five rounds per hour, it is seen that the limit of its life would be reached in less than two hours. With the 14-inch gun, the life of the gun before its accuracy would show impairment would be about 240 rounds, corresponding to about six and a-half hours of continuous firing at the rate at which it is thought it can be fired. The weight of the new gun will depend greatly upon the manner in which it is built. It is yet to be decided whether the guns shall be of the ordinary built-up forged-steel type or shall be of the more modern wire-wound construction. If wire-wound, the weight will be about 110,500 pounds, while the built-up type will weight 12,500 pounds more. The present standard gun weighs about 1,500 pounds more than the average for the new gun. They will be mounted on the disappearing carriage, and will be installed wherever needed.—"Scientific American."

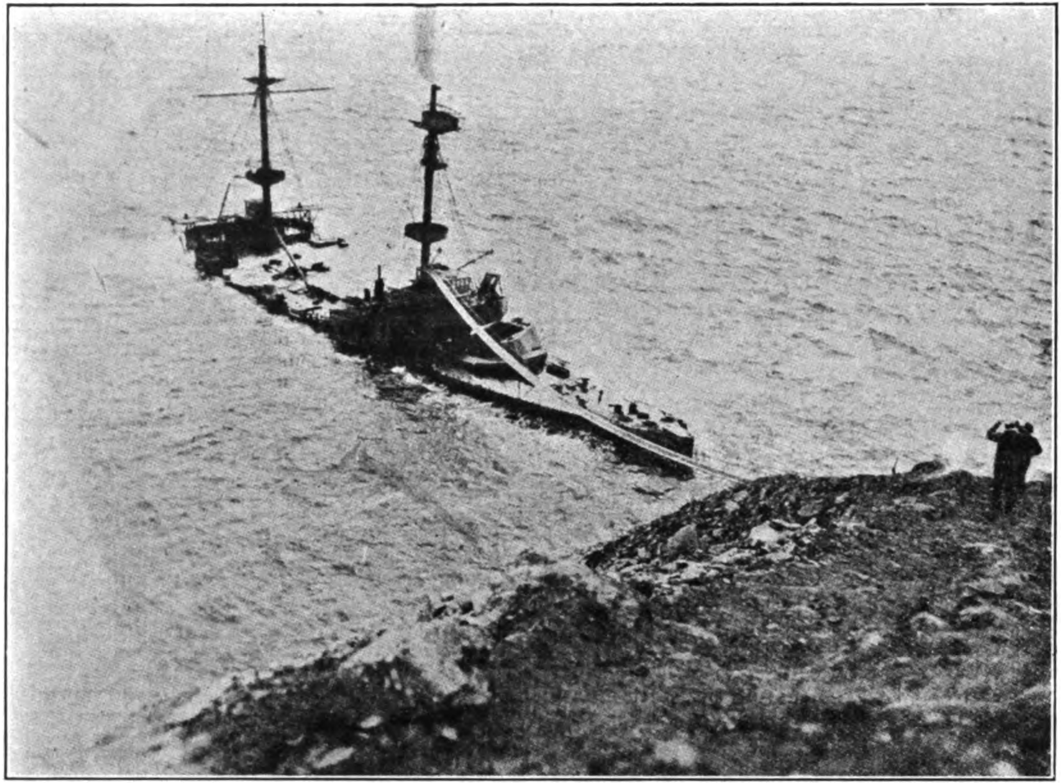
BREAKING UP THE ILL-FATED BRITISH BATTLESHIP *MONTAGU.*

By Harold J. Shepstone.

The first-class British battleship *Montagu,* which went ashore on the southwest corner of Lundy Island, in the British Channel, in a dense fog on May 30, 1906, is now being broken up piecemeal by the shipbreakers, and gradually the huge warship is disappearing.

The gallant attempt made to salve the vessel, although a complete failure, was nevertheless a daring piece of work deserving of the highest praise. For months the Liverpool Salvalge Association, and the most skilled workmen from the British dockyards, with the most up-to-date tackle at their disposal, endeavored to lighten the ship and get her off the rocks, but without success. The whole history of the operations was one steady and persistent fight against unsurmountable difficulties.

Twenty-four hours after the vessel grounded, practically every compartment in the ship was full of water. The cap-



THE WRECK OF THE "MONTAGU" AS IT APPEARS TODAY OFF LUNDY ISLAND.

stan-engine room forward, the compartments under the forward 12-inch turret, all the boiler rooms, the starboard engine room, and the steering compartments aft, were open to the sea, the water rising and falling with the tide. The port engine room was tight, but had to be flooded by opening the sluice valve between the two engine rooms to prevent the ship taking a heavy list. At high tide the water rose about two feet above the upper deck at its lowest part, while at low tide the water fell to about seven feet below the usual waterline of the ship.

The vessel rested on a rocky bottom, the general surface being fairly level, but with many pinnacle rocks. It was discovered by divers that the damage to the bottom was very serious. A large rock penetrated about ten feet into the ship under the capstan engine forward. Along the starboard side the bilge keel was more or less carried away, and several large holes were discovered. The port propeller, A bracket, and shaft had disappeared completely; the starboard A bracket was cracked, and one blade of the starboard propeller was carried away, and the lower parts of the sternpost and rudder were broken off. To repair the vessel and pump out the water so that she could be refloated was the task the British Admiralty assigned to Admiral Wilson, and every possible assistance was rendered him in his work.

After a complete examination of the ship it was decided to pump the water out of the vessel as far down as possible by pumps driven by compressed air. Before this was done the battleship was lightened by removing the 6-inch and smaller guns, torpedo nets, chain cables, etc. Owing to the surrounding rocks and shallow water, only lighters or very small vessels were able to approach to the wreck. This added greatly to the difficulties of the operations, as in most cases heavy weights, such as boilers, etc., had to be first hoisted into lighters. One of these craft, containing four 6-inch guns and several torpedo nets, in being towed was carried by the current over the rocks and foundered. She was, however, successfully salved some weeks later.

Within ten days from the disaster the following centrifugal

pumps were in action: One 12-inch, three 10-inch, three 8-inch, one 5-inch, and one 3-inch, and an additional 12-inch pump was available on board one of the salvage boats supplied with steam from her own boiler. Three more 10-inch pumps and two 8-inch pumps were also in course of erection. A week later seven more pumps had been fitted up on board. They had a total pumping capacity of 8,600 tons per hour. Their object was to command the different compartments in which they were placed, and generally clear the water down to about two feet above the platform deck. Owing to the size of the rose boxes they could not clear the water closer to the deck, and efforts were then directed to find compartments below this level which were tight, and which could be used as wells for pumping suctions. In the fore end of the ship this scheme had to be given us as hopeless, though it was partially successful in the after part of the vessel. Here a U-shaped pocket or well was lowered through the hatch of the submerged torpedo room, and guided into place under the deck and shored up by divers.

Although it was found that in many ways the minute subdivision of the ship helped the salvage work, it made the work of pumping the ship clear of water more difficult. To prevent the vessel bumping at high water it was generally necessary to let her fill up with water then, and arrangements had to be made to pump out the various compartments rapidly, say in about eight hours. It was not practicable to provide special pumps or suctions for all the minor compartments, and time did not permit of the suction pipes being moved from one compartment to another. Drain holes had therefore to be cut —mostly under water—to enable the smaller compartments to drain into the larger ones, where pump suctions were fitted.

The next task was the covering over of the hatches to the engine and boiler rooms at the elevation of the main deck by plates. Then a number of camels, or tanks, were fixed along the side of certain portions of the ship. By experiments it was estimated that to float the vessel safely, a plant capable of producing 6,000 cubic feet of air per minute would be required

to drive the water out of the ship. It was obtained, but failed to respond to the gallant efforts of the salvage operators.

On August 6 last the British Admiralty, after nearly fifteen months' strenuous efforts to refloat the warship, decided that the task was hopeless and abandoned it. Before doing so the vessel's heavy guns, a large quantity of her armor plates, boilers, machinery, and chain fittings, were removed. The wreck was then purchased by a South Wales syndicate for a lump sum, who hope to secure the valuable steel and iron composing the hull.

The syndicate at once established quarters on the island, where they have a staff of some fifty experienced workmen and a couple of small steamers and lighters. Their engineers have thrown an aerial footway, over 500 yards in length, from the top of Lundy's precipitous cliffs to the roof of the chart house on the wreck. Down this footway the shipbreakers pass to and from their work. The main deck of the ship is always awash at high tide; therefore work can only be carried on for a short time daily. When weather permits, lighters are brought alongside the battleship, immediately below the footway, and piece by piece huge sections of the armor plating are being removed from the huge carcass and transferred to the shore. The amount of work which has been done on the hull will be gathered from the huge "bite" which has been taken out of the bows on the port or seaward side, but it will be many months yet before the *Montagu* has been completely broken up.

One result of this wreck will probably be that the British Admiralty will establish a salvage corps of its own. Hitherto it has had to depend upon outside assistance, whenever any of its vessels have met with serious disaster. So far the British Admiralty have not stated what the salvage operations have cost, but it is estimated to amount to no less than $500,000, while as to the ship herself, she was built so recently as 1903, was of 14,000 tons, and cost the British nation close upon $5,000,000. What the syndicate paid for the abandoned wreck has not been disclosed, but probably only a few thousand dollars."—"Scientific American."

MATERIALS FOR THE CONTROL OF SUPERHEATED STEAM.

A paper presented at the Indianapolis meeting (May, 1907) of the American Society of Mechanical Engineers, by M. W. KELLOGG, '94, M. E.

GENERAL.

Since the introduction of superheated steam as an important factor in economy in stationary power-plant operation, the question of what type of material is best for the proper controlling of the resulting high temperatures has caused a great deal of investigation and interest.

In the following discussion of materials some reasons will be given which are the results of experience and test, and other facts which we have accumulated from reliable sources will be shown. This article treats particularly of what might be called in a general way piping systems, which systems are made up of pipe, fittings, valves and the necessary details connected therewith, such as joints, gaskets, etc., and are here taken up separately.

PIPE.

There can be little question as to the matter of pipe except quality. Of course, welded wrought-iron or steel pipe is successful, but the difference in the quality of pipe under different conditions is very material. As in nearly all instances in a superheated-steam station the old-fashioned screwed joint is not satisfactory, it is necessary to do what is termed "work" the pipe—that is, weld, van stone, etc.—to make either a welded, van-stone, or other joint of the same general description.

The accompanying cut is what is known as a "van-stone joint." For this work, the pipe made from open-hearth steel is a great deal the best for manufacturing reasons, because it can be properly "worked," there being less carbon and the quality being much more uniform. Bessemer-steel pipe will very often act in a satisfactory manner, but one is never sure that Bessemer will run even, and, therefore, troubles may result.

It is practically impossible to "work" wrought-iron pipe. In making what is known as a "van-stone joint," the pipe is

nearly sure to split very badly, not only at the weld but all around its outer circumference.

Nearly opposite qualities from those used for getting good results from "working" pipe are required for threading. A good quality of wrought iron will cut and thread more easily with standard pipe machines and standard dies than a steel pipe, and a Bessemer-steel pipe will thread much more easily with standard dies than open-hearth steel will.

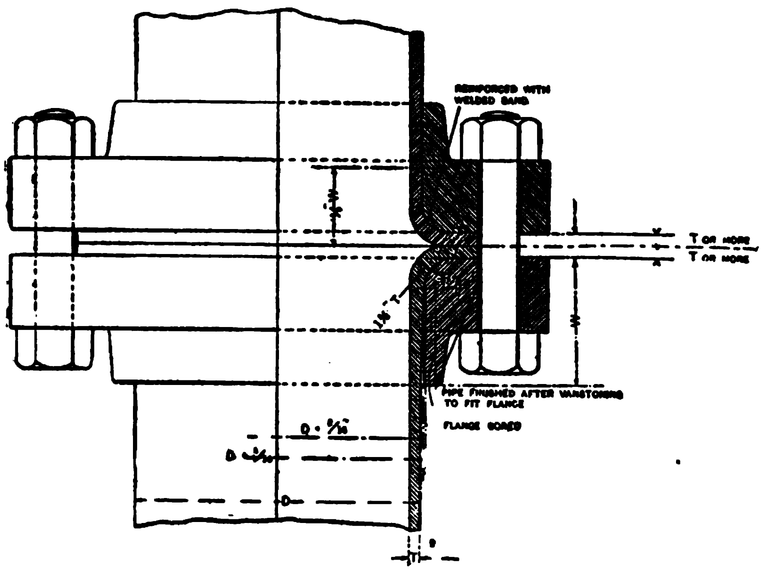


Fig. 1.—VAN STONE JOINT.

A great many manufacturers have difficulties in threading open-hearth steel pipe, for the reason that they set the dies exactly the same as if they were cutting other qualities. This causes ripping of threads, etc. The die in a pipe machine should be set at a greater angle, with the radius of the pipe passing through the point of contact of the die for soft steel than it would be for other kinds, and this in itself will very often eliminate great troubles in this line. The question of lubrication, etc., is also important in this particular.

The ordinary commercial pipe will stand more pressure than

the average person believes. A standard one-inch piece of welded pipe will usually not break under 1,600 pounds per square inch hydraulic pressure. Full-weight pipe I believe to be perfectly suitable for any temperature and any working pressure up to 225 or 250 pounds, as long as it is not thinned at any point by cutting and threading.

FITTINGS.

The designs of fittings as generally manufactured for the different purposes are in a general way very satisfactory, with the one exception that very few manufacturers on their standard articles include what is known as the "long fillet" between the body of the fitting and the flange. This is a very desirable point, owing to the fact that at this place there is the greatest strain from shrinkage in the molds, which also tends to develop porous spots. Most large users of this type of material have learned this thoroughly and design their fittings specially, the chief difference in their design from that of the general manufacturer merely covering this point. The quality of the material in fittings, however, is a very important thing in connection with superheated steam.

The latest practice is to do away with fittings entirely on high-pressure steam lines and put what are known as "nozzles" on the piping itself. This is accomplished by welding wrought-steel pipe on the side of another section, so as to accomplish the same result as a fitting. In this way rolled or cast-steel flanges and a van-stone or welded joint can be used. This method has three distinct advantages, to wit:

(*a*) The quality of the metal used, for reasons explained hereafter when the subject of the effect of heat on metals is taken up.

(*b*) The lightening of the entire work.

(*c*) The doing away with a great many joints.

As a general average, at least 50 per cent. of the joints can be left out, and sometimes this proportion runs up as high as 60 or 70 per cent., according to the layout of the system. If this method is employed substantial welds must be made, not

only to stand the pressures required but also the strains; this is accomplished successfully in Germany, England and the United States.

VALVES.

It is important to have a good design of valve. I believe that nearly all of the designs made by the good manufacturers are entirely suitable, such as a broken or solid wedge valve of the ordinary type, under the condition that all machine work is done thoroughly and the quality of metal used is satisfactory for the purpose intended.

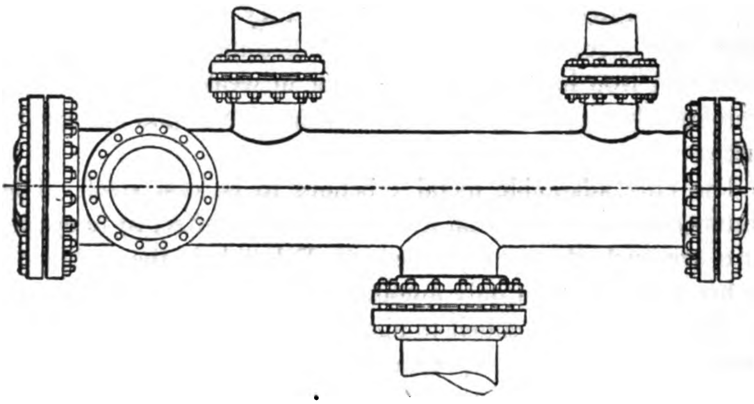


Fig. 2.—Welded Work.

It may be interesting to note here the effect of a large range of temperatures on a short piece of steel. By calculation, a piece of steel six inches long, heated 500 degrees, will expand 0.019 inch. This figure is put down to show how variations in the coefficient of expansion of metals by heat have a large effect on the permanency of a valve staying tight, and it can readily be seen that a small part of the distance given is sufficient to cause trouble.

METALS.

I find that different authorities vary slightly in their statement as to what temperatures different metals will stand with good results. German authorities state that cast iron

should not be used above 480 degrees Fahrenheit. Other authorities allow us to go as high as 575 degrees Fahrenheit. Above these temperatures in cast iron the limit of elasticity is reached with a pressure varying from 140 to 175 pounds. Under such conditions the material is strained and does not resume its former shape, and eventually shows surface cracks, which continue to grow until it lets go. These temperatures and pressures also lead in time to a shrinkage of all parts, and to a structural alteration of the metal, which results in leakages in valves at the seatings. Therefore it would seem that iron castings are unsuitable for both fittings and valves to be used in any superheated-steam work. While they may last for some time, after a few years' use the metal becomes very weak, and some cast iron has reached the point in weakness where if it were merely tapped lightly with a hammer it would break into pieces.

The only adaptable metal I believe to be cast steel. The results of tests by Bach on this metal for the effect of temperature are such that at 572 degrees Fahrenheit the reduction in breaking strength only amounts to about 1.1 per cent. and at 752 degrees Fahrenheit to about 7.8 per cent. Therefore it seems that this metal is practically capable of withstanding all pressures and temperatures up to at least 800 degrees Fahrenheit, without showing any appreciable weakness.

The influence of high temperatures on bronze, etc., is very material. At ordinary temperatures this metal has a breaking strength of about 34,100 pounds per square inch and an elongation of 36 per cent. At 572 degrees Fahrenheit the breaking strength falls to about 19,500 pounds per square inch and the elongation to 11.5 per cent. At 662 degrees Fahrenheit, which is quite a common temperature, as the steam leaves the superheaters, the breaking strength of bronze only amounts to 12,200 pounds per square inch and the elongation at the breaking point is only approximately 1¾ per cent. This seems to eliminate entirely brass or bronze of ordinary composition for use with highly superheated steam.

The effect of temperature on nickel is very similar to that

on cast steel, and in consequence this material is very suitable for use in connection with highly superheated steam. Bach recommends that bronze alloys be done away with for use on steam lines above a temperature of about 390 degrees Fahrenheit. Even neglecting the special quality of nickel seatings, on account of the great toughness of this metal and the methods which can be used for securing rings of this substance to the valves and conical surfaces, it has the special advantage of having the coefficient of contraction and expansion with temperature almost exactly the same as that of cast steel, so that no slackness of the rings occurs and the valves remain absolutely steam tight. There are instances in which valves constructed with nickel seatings have been satisfactorily used with steam temperatures up as high as 932 degrees Fahrenheit.

Seats, discs and bushings made of brass or plain bronze do not retain their shape. For spindles on superheated-steam work I strongly recommend nickel-steel, which holds its shape and does not deteriorate with high temperatures. Seatings in valves should not only be screwed in but also pinned in addition, using a fine thread which is very long, to give a tight joint. Seats should also have a flange on the top that makes a joint with the body when screwed down, which prevents the tendency to leak through.

JOINTS.

I think it is generally acknowledged that the old-fashioned screw joint, no matter how well made, would not be suitable for superheated steam work. This leaves for discussion two general types, viz: welded joints, and what are generally known as van-stone or climax joints; that is, any joint where the pipe is turned over the face of the flanges.

In welding a flange on a piece of pipe great care must be taken to see that the weld is perfect, because of the unequal thicknesses of the metal to be so welded. If the weld is thoroughly made this type of joint is very good, although for erection purposes, owing to the fact that the flanges cannot swivel, it does not equal the turned-over joint, as mentioned

above. The manufacturing expenses in making a welded joint are also much more for the same type of work accomplished, on account of the necessity of doing all finishing work after all rough work, such as welding and bending, has been completed. Therefore the cost of welded joints is greater, not only for the work done but because of the increased expense in finishing on account of the necessity of employing methods different from those where the flanges, etc., were all finished before the joints were made, as is possible on the turned-over joint mentioned above.

In regard to the turned-over or van-stone joint, the quality of its manufacture seems to us the most important feature. This joint can be made in a careless way where the pipe is in no way thickened up and only faced on the front. A joint of this kind does not give good results, principally for two reasons:

(*a*) The thinness of the metal on the turned-over portion; and

(*b*) On account of the recesses left between the back of the pipe on the turned-over portion and the flange, owing to the pipe not being finished at this point.

The writer believes, however, that if this point is properly made it is equal to the welded joint as a manufactured article and superior to the welded joint as an article for erection.

To have this type well made the pipe on the end should be thickened up in an amount so great that after the joint is turned over there will be enough metal left to face the turned-over portion on the front, on the outer edge and on the back. We of course take for granted that the flanges are finished on the front. After the work above mentioned is done, the pipe should be as thick on the turned-over portion as the original thickness, or very close to it. Increasing the thickness of the pipe on the end before going through the operation is done in several ways. In a general way, I consider any of the methods satisfactory. The point made of facing the turned-over portion of the pipe on the back is an exceedingly important one, much more so than most people seem to

realize. I have known instances where it has been found impossible to make a ground joint, for no other reason.

In reference to making up a joint, I believe that the face of all flanges or pipe where a joint should be made ought to be given a fine tool finish and have the face level, and that then a gasket of some description should be used. A perfectly made ground joint is a good thing but it is very expensive, and it is hard to get the average contractor to furnish it in a perfectly workmanlike manner. Also, after it is so done, it is liable not to stay tight, on account of the tremendous expansion and contraction causing such strains that the joints are liable to open up, particularly when the pressure is taken off the plant. The simple expansion and contraction on the bolts that make up a joint would cause this.

GASKETS.

There are large numbers of gaskets manufactured of all types and descriptions. It is very hard to take up this subject and be fair to each of the manufacturers, for the reason that practically no one can and has had experience with every type made, so as to judge for himself, and hearsay would lead us to suppose that all of them are at one time perfect and at other times useless.

I have used a great many different types of gaskets, however, and have obtained the best results with a corrugated soft Swedish steel gasket with "Smooth-on" applied, and with the McKim gasket, which is of copper or bronze surrounding asbestos. The ordinary corrugated copper gasket is a very popular make and has been used a great deal. On superheated steam, usually sad results follow. There seems to be some peculiar action that causes this, as on superheated-steam lines a corrugated copper gasket will in time pit out in some part of the flange nearly through the entire gasket. I have heard a great many reasons given for the cause, such as electrical action, disintegration, etc.

The wear of a gasket depends largely on the method of pulling up bolts on flanges. In fact I believe that a great many

troubles have occurred because of imperfect erection. If joints are pulled up entirely on one side and left loose on the other, and then taken up on that side, trouble with the gasket is almost certain. The bolts should be taken up gradually all around the flange. The experience of the erecting crews on high-class superheated-steam lines is an exceedingly important thing. The average steam fitter is not suited to this type of work. He has had experience with lower pressures and less important tasks, and after a piece of work is erected by him it is customary to find a great many leaks which are usually eradicated only after the whole joint has been broken and properly repaired. All these troubles can be eliminated by using only steam fitters experienced in the type of work under consideration.

Referring again to metals, cast iron and bronze, after being very severely heated and cooled off, as they naturally are in a superheated-steam line, do not seem to retain their original shape. I know of places where gate valves, after being installed in superheated-steam lines and being in service for approximately two years, are one-half inch larger face to face than when they were installed. There have also been accidents due to the fact that cast-iron flanges on wrought-iron pipe have expanded, as a total, and never returned to their former shape, thereby causing weakness and breaks.

Summing up, I believe good practice in steam lines of this type is to do away with cast iron and bronze throughout and with all fittings, using welded wrought-steel outlets, as illustrated in Figure 2, and cast-steel bodied valves with nickel-bronze seats and nickel-steel stems.

MARINE TURBINE LUBRICATION.

Read before the Institute of Marine Engineers by A. H. MATHER.

One of the results of the progress made in the design and construction of marine steam turbines, and the consequent increase in the number of vessels fitted with turbine machinery, has been the acquirement of increased knowledge as to the

problems arising in the management and running of this class of machinery. Amongst others, the problem of lubrication has forced itself upon the notice of engineers with the usual insistance which lubrication, or the lack of it, always exhibits whenever it gets the opportunity of showing itself. Marine turbines are fitted with a system of forced lubrication, which provides for a good flow of oil through the bearings, but which might be considerably improved by attention to some matters of detail considered from the standpoint of treating the oil in such a way as to ensure the best results. I have examined a number of sets of turbine machinery, and I find that while in the majority of cases every attention has been given to the mechanical details, very little thought has been given to securing the best conditions of work for the oil itself; and it has, therefore, occurred to me that a short paper dealing with this side of the subject would be both of interest and value to the members of the Institute. I do not propose to deal with the subject of turbine lubrication from any other than the mechanical point of view, and have, therefore, confined myself to the consideration of one or two matters of detail in the arrangement and fitting up of the storage tanks and pipes, attention to which would ensure that the oil in use would receive the best treatment, and would lead to a marked improvement in the running of the jobs.

STANDARD OF OIL REQUIRED.

Before entering into the consideration of these details, it might be as well just to enumerate the qualities required in an oil for this class of work:

1. It must be entirely free from saponifiable matter.

2. It must separate rapidly from any water picked up in its passage through the system.

3. It must retain its lubricating value when heated to a fairly high temperature.

4. It must retain its nature in continuous use for long periods.

In forced-lubrication systems it is essential to keep the oil

fully up to its standard, and to ensure this being done the two most important points to be considered are:

1st. Its complete separation from water picked up in its passage through the system.

2d. Its thorough filtration from grit or material substances.

FILTRATION AND SEPARATION SYSTEM.

The necessary installation on board a turbine ship consists of a tank or tanks, a filter and a force pump, and it is in regard to the arrangement of these that I propose to make a few suggestions which should result in the marked improvement in the lubrication and in economy of oil. For the purposes of this paper I have prepared a special diagram, which diagram, I may say, is based simply upon a knowledge of the properties of oil, and has been prepared with the assistance and approval of a chemist unrivalled in his intimate knowledge of oils and their proper treatment and uses. It has been recently submitted to all the prominent engineering firms throughout the Kingdom, and its principles have been adopted in a number of cases with satisfactory results.

A complete system for the filtration and separation of oil and water should consist of two storage or settling tanks, two filters, an oil cooler and a force pump.

DETAILS OF THE SCHEME.

The two tanks, which should be placed low down in the engine room to allow all the oil in the bearings and pipes to drain freely back by gravity, should each be of sufficient size to contain enough oil to fill the whole system outside of the tank, and having also an ample margin or capacity in the tank to keep a reserve of oil above the pump suction pipe. The tanks should be tilted or a pocket should be made so that the drain cock is at least two inches lower than any other part of the tank. The drain cock should be of large size, and means should also be provided to show clearly the amount of water in the tank. For this purpose a glass-sided box placed at the lower end of the tank and having adequate protection for the

glass sides is preferable to the ordinary gauge glass, as it gives a clearer and more reliable view. An air pipe should be fitted at the highest point in the tank, and led as high as necessary. The oil-return pipe from the bearings should enter the tank as near to the bottom as possible, and it is recommended that the oil be allowed to return in its heated condition in order to facilitate its separation from the water. Oil in a heated condition will also allow suspended matter to drop to the bottom more easily. In cases where a large proportion of water finds its way into the oil, as, for instance, with ordinary reciprocating marine engines fitted with forced lubrication, where it is practically impossible to prevent a considerable quantity of water from the glands, etc., falling into the crank pits and mixing with the oil, a small heater should be provided in the return pipe to raise the temperature of the oil to about 200 degrees Fahrenheit. This will result in the immediate separation of all water and foreign matter as soon as the mixture find its way into the tank, the oil rising rapidly to the top and the residue falling to the bottom. To obtain the best results this heater should be fitted to all forced-lubrication installations, including turbine sets, although no doubt it may be urged that it is adding to the cost of the plant and increasing the space occupied where the amount available is generally very limited. The pump suction should be placed as high as possible, considering the amount of oil needed in the system, so as to allow ample space for the accumulation of water and extraneous matter to remain at the bottom without being drawn through, the opening of this pipe to be directed upwards if possible. The net storage capacity of the tank is of course its capacity above this pipe opening. In the majority of installations I have examined, the inlet and outlet pipes are placed in positions the reverse of those which are here recommended, that is to say, the inlet or return pipe is led into the top of the tank, thus allowing the returning oil to fall the whole depth, thoroughly disturbing the contents and effectually preventing the separation of any of the water, and, as if this were not bad enough, the suction pipe is taken from the bot-

tom of the tank with, in some cases, an internal bend to carry it down to within an inch or so of the lowest point. This latter is done, of course, with the idea of utilizing the full capacity of the tank, but it results in keeping the turbine supplied with a mixture of oil and water. Two tanks should be supplied, and these should be used alternately at convenient intervals of perhaps two or three days, for, although the arrangements of pipes suggested allows for separation during the process of circulation, further separation is necessary to clear the oil, which can only be done by allowing it sufficient time to rest and obtain a thorough settlement.

THE FILTERS.

Two filters should be supplied to allow of one being cleaned out while the other is at work. They should consist of three or four separate layers of brass-wire gauze of, say, 24 mesh to the inch, the last or uppermost one consisting of two sheets of gauze with a sheet of cheese cloth between them. The oil should enter the filter at a point some distance above the bottom but below the layers of gauze. As the oil rises through the filter any particles of grit, dirt, etc., that may be drawn from the tank are retained in the lower part of the filter and tend to fall away from the filtering surface, which will thus remain clear for a longer period than would be the case if the oil traveled downwards, in which event all foreign matter would be retained on the filtering surface. In addition, this system of filtration would also assist in the further separation of the oil and water. For this reason sufficient capacity should be provided in the lower part of the filter, below the inlet, to collect the extraneous matter and prevent it from passing back through the pipe into the tank. A cock should be provided at the lowest point of the filter to drain this chamber.

Stop valves or cocks should be provided in the oil pipes for changing the tanks over, and it would be convenient to be able to use either filter in connection with either tank, as

one filter might, on occasion, require cleaning out when it is not desired to change the tanks.

After being filtered the oil should pass through a cooler having sufficient surface and water circulation to reduce the temperature to normal. This cooler may be of any convenient design, provided that it allows free and ample passage for the oil, and does not allow the cooling water to find its way into the oil. Probably the cheapest form to be efficient would be a series of tubes with expanded ends in a casing through which the sea water can be pumped, the tubes being arranged so that the oil passing inside them is made to travel two or three times through the water. I have seen a cooling coil fitted in the bottom of the storage tank, but this has the objection that if any leaky joints occur the water gets direct access to the oil. Another method in use is a water jacket round the tank, but this is not very efficient, as only a portion of the oil comes into contact with the cooling surface, and, in the particular instance of which I am thinking, the jacket is only supplied by a half-inch pipe, giving a very small circulation, although the tank holds probably 120 gallons. A cooler of this type should be so constructed that the cold water enters at the points where the oil leaves it before going to the bearings.

The oil pump should preferably be placed between the filter and the cooler, as there is then less likelihood of water obtaining access to the oil through leaky joints in the cooler, any leak occurring allowing a leakage of oil outwards instead of water inwards.

OTHER DETAILS.

Leaving this part of the installation, one or two matters connected with smaller details are worth referring to. The oil is distributed from the pump to each bearing, a pressure gauge being provided to show the pressure at the pump discharge. Means should be provided to show whether each separate bearing is obtaining its proper supply of oil, and also to regulate the quantity supplied to each bearing. Sight-feed

cups are fitted to a number of jobs, but some of these are so arranged that they do away with the oil pressure, the oil entering the top of a cylindrical glass cup and falling visibly into a funnel placed lower down in the cup. The pressure is, of course, lost where the oil leaves the upper pipe, and the bearing is not under forced lubrication, the pump, to all intents and purposes, simply becoming a circulating pump. A better plan is to have a similar fitting placed on the drain from the bearing which will show the amount of oil which has actually passed through the bearings while permitting the pressure to be carried right up to the working surface.

One part of the turbine which is capable of giving, and has given, very serious trouble, is the steam-packed gland where the shaft passes out of the turbine casing. The packing rings are a very close fit, and really require better lubrication than that provided by the wet steam, but the admission of oil here is a serious matter, as it simply passes straight into the turbine and through into the feed water. It seems to be a necessary precaution, however, to give the glands a little oil when working into port to prevent seizing or rusting of the shaft to the gland when the turbine is not at work, and I understand that in some cases a branch has been provided in the small pipe supplying steam to the gland to admit of a small quantity of oil being introduced when required; but a connection of this kind is a risky thing, as it would be very easy for a careless or thoughtless man to put in sufficient oil to cause boiler troubles.

In conclusion I may say that in presenting this short paper to the Institute I have done so considering that a few notes on a side issue in connection with what is probably destined to become the premier method of ship propulsion would be of general interest to all, and might be of service to some who have either the building or running of this class of machinery under their control, and if it is of use in showing where trouble may arise in the lubrication of turbine machinery, or in offering a suggestion which may lead to an improvement in the

efficiency and economy of the lubrication system generally, I shall feel satisfied that the few ideas which have suggested themselves to me in the course of my observations and experience have been worthy of placing on record.—"Page's Weekly."

SHIPS.

AUSTRO-HUNGARY.

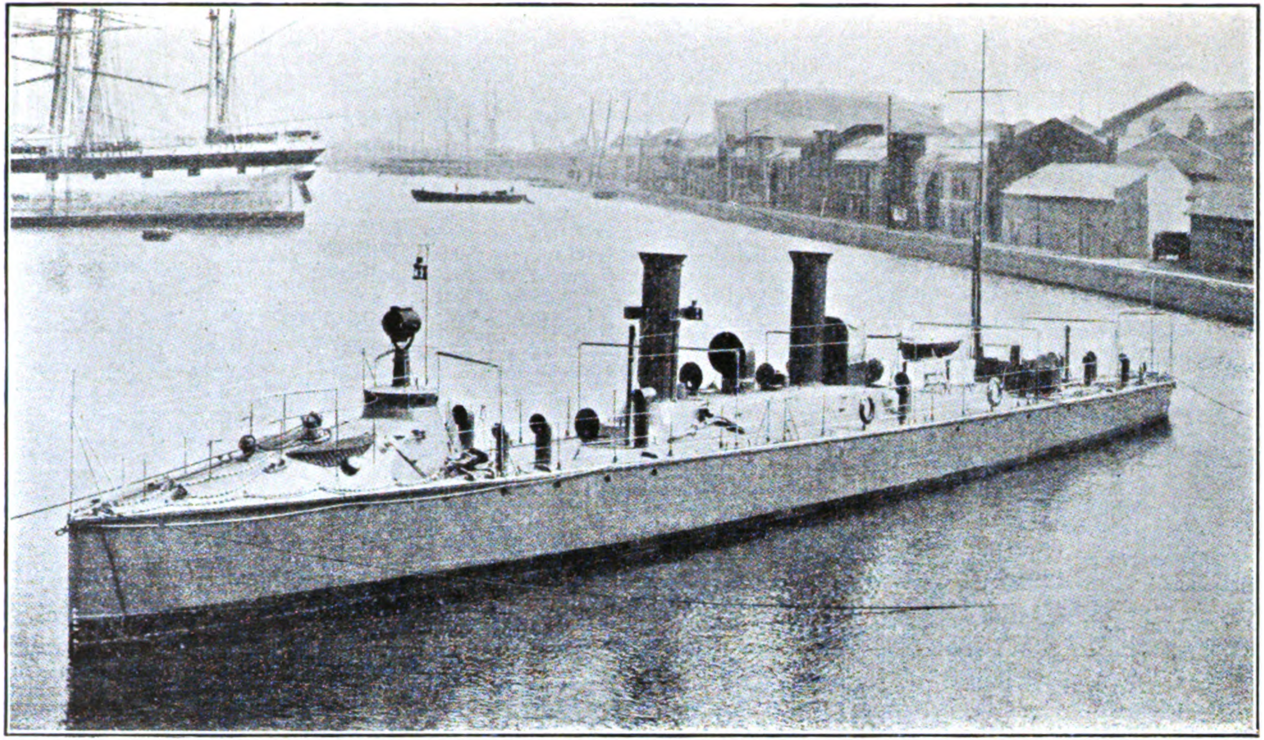
The Austro-Hungarian Government have contracted with Messrs. Yarrow & Company (Limited), of Poplar and Glasgow, for the construction of two exceptionally high-speed shallow-draught gunboats, propelled by internal-combustion engines. These vessels are to be in all essential particulars similar to *Mercury II*, which was purchased by the British Admiralty last year, and on board which, it will be remembered, the King and Queen took a trip during Cowes week.

BRAZIL.

Torpedo Boat.—We give an illustration of the first-class torpedo boat *Govaz,* which has just been built by Yarrow & Co., Limited, of London and Glasgow, for the Brazilian Government. The vessel is 152 feet 6 inches in length by 15 feet 3 inches beam, and has a displacement of about 150 tons in service condition with bunkers full. This type of topedo boat is similar to those built in recent years by Messrs. Yarrow & Co., for the Austrian, Chilian and Dutch Governments. The armament consists of two 47-mm. quick-firing guns, and two 45-cm. revolving torpedo tubes, which are to be placed on board when the vessel arrives at Rio de Janeiro.

The general internal arrangements and crew spaces are the same as is usual in this class of boat; the seamen and firemen being berthed forward, and the officers and warrant officers aft.

The machinery consists of two "Yarrow" water-tube boilers, fitted with forced draft and fed by independent feed engines. The main propelling machinery is on the Parsons' system of turbines combined with triple-expansion reciprocating engines. The reciprocating engine is used for cruising purposes, and is



First-Class Torpedo Boat for the Brazilian Government.

capable of driving the vessel at a speed of 12 knots on a very low coal consumption, in which respect turbines when working far below their full power are uneconomical.

When high speeds are required the turbines are used in conjunction with this engine, which is arranged for the large variation of revolutions necessarily entailed.

The turbine machinery consists of one high-pressure and one low-pressure turbine, with the usual independent forced-lubrication pump, air pump and circulating pump.

The three-hour official full-speed trial was run on the 30th of May, in the Estuary of the Thames, in the presence of Rear-Admiral Joao Justino de Proença, Chief of the Brazilian Naval Commission, Captain Josè Thomaz Machado Portella, Chief of Naval Construction, and Captain Bartholomeo Francisco de Souza e Silva, Chief of the Machinery Section; and during this trial six runs were made over the Admiralty measured mile on the Maplin Sands, with the following results:

Steam pressure in boilers, 230 pounds per square inch; in cruising engine, 207 pounds per square inch; at H.P. turbine, 220 pounds per square inch; at L.P. turbine, 29.6 pounds per square inch; vacuum in condenser, 27 inches; air pressure in stokehold, 1.8 inches.

For the three hours the mean speed was 26.493 knots.

Following the above, a consumption trial was made in the presence of Captain Portella and Captain Bartholomeo da Silva, when a mean speed of 11.277 knots per hour was obtained, with 150 pounds steam pressure in the cruising engine, the turbines being shut off, and the coal consumption under these conditions showed 56 knots run per ton of coal burned.

The above figures fully justify Messrs. Yarrow's decision in adopting the combination of reciprocating engines and turbines first suggested to them by Captain Soliani, of Genoa, in view of the fact that turbines have not yet been found economical at low power, even when a special cruising turbine has been fitted. Another difficulty avoided by this system is that of going astern, the reciprocating engine being fitted with reversing gear, as usual, and no astern turbine is, in consequence, nec-

essary. It may, therefore, be said that the cruising engine takes the place of a cruising turbine and an astern turbine, and, of course, weighs considerably less, and in addition has the great advantage of being an engine the design of which is familiar to every engineer. No difficulty is met with in arranging the reciprocating engine to run at the revolutions necessary for full speed, as the slip of its propeller reduces as the speed of the vessel increases, so that although the speed of the boat rises, say, from 11 to 26 knots, the speed of the reciprocating engine only rises from about 340 revolutions to 540 revolutions per minute. Hence, though the speed is considerably more than doubled, the revolutions of the cruising engine only increase some 60 per cent.—"Engineering."

Customs Cruiser for Brazil.—A twin-screw cruiser yacht named the *Amapa* has recently been built by Messrs. John I. Thornycroft & Co., Limited, at Woolston Works, Southampton, for the Customs service of the Brazilian Government. She is constructed of steel to Lloyds "100 A. I." class, and on the official trial, which was carried out on May 28th last, proved herself, so we are informed, an excellent seaboat, and maintained a speed of 12½ miles, with natural draft. This was, it may be added, half a mile in excess of the contract speed. A speed of 15¾ miles was attained with forced draft, this being three-quarters of a mile above the contract speed. Her principal dimensions are: Length between perpendiculars, 130 feet; beam, molded, 17 feet; depth, molded to main deck, 9 feet 9 inches. A chart house with a flying bridge over it is placed at the aft end of forecastle deck, on which are situated the searchlight, steam-steering gear, compass and telegraph. A three-pounder quick-firing gun is mounted on the forecastle head, and a combined steam and hand windlass is also provided. The accommodation for the officers and guests is under the raised quarter deck aft, while the petty officers and crew are berthed forward. The galley is on the main deck. Stores, fresh-water tanks, magazines and ballast tanks are arranged under the cabin flats.

The vessel is lighted throughout by electricity, and fresh air

is circulated through the cabins by means of electric centrifugal fans. Three boats are carried. One of these is a 25-foot lifeboat, another a 25-foot cutter, and the third a 12-foot dinghy. Two Downton pumps are fitted for clearing the bilges and for washing deck and fire purposes.

The machinery consists of two sets of triple-compound surface-condensing engines working at 300 revolutions per minute, having cylinders of 9 inches, 13 inches and 20½ inches diameter, and 11 inches stroke. Steam is supplied from a water-tube boiler working under forced draft on the closed-stokehold principle, the working steam pressure being 180 pounds per square inch.

ENGLAND.

The New Ocean-Going Destroyers.—The first five ocean-going destroyers of the Tribal class—so appropriately named *Afridi, Cossack, Ghurka, Mohawk* and *Tartar,* respectively—are approaching completion.

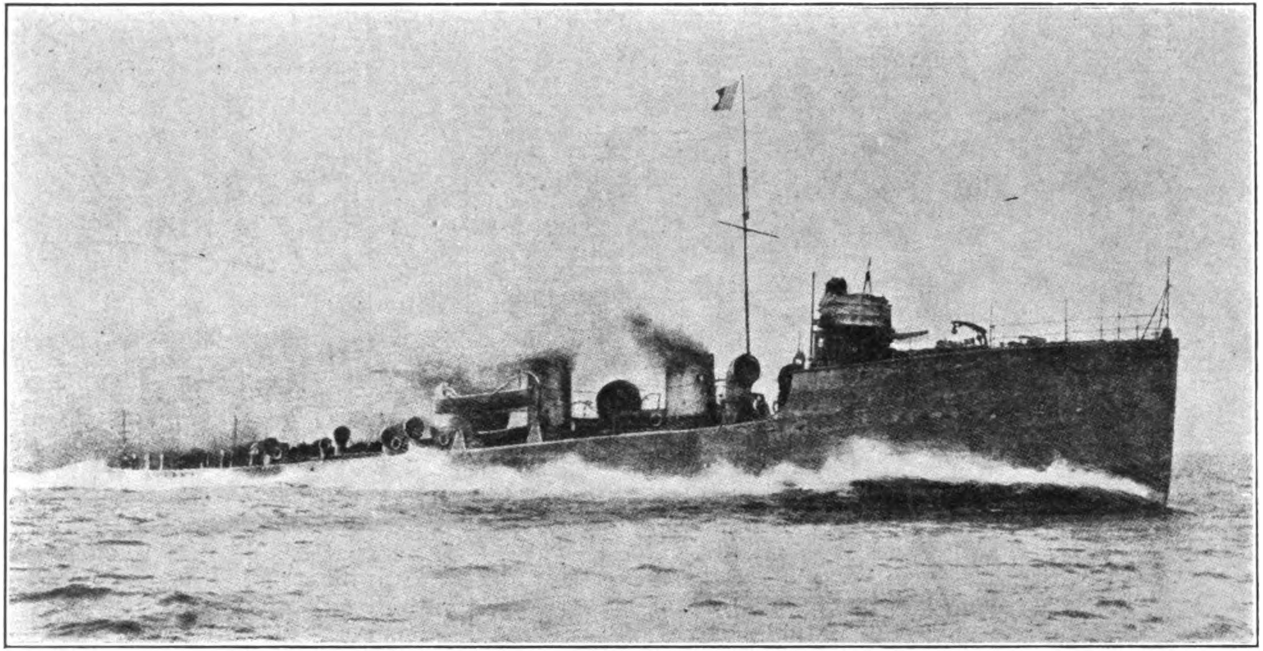
The *Cossack* is 270 feet in length, 26 feet in width, and has a depth in hold of 15 feet 5 inches. She is fitted with Parsons turbine engines, which have been constructed by the builders at their Birkenhead Works. They are arranged on three shafts, the high-pressure working the center screw, and the low-pressure and cruising turbines driving the wing propellers. Steam is supplied by five water-tube boilers of the Express straight-tube type, working at a pressure of 220 pounds per square inch, and fired with oil fuel by sprayers of the Admiralty type. Her armament consists of three 12-pounder quick-firing guns, one on each side of the forecastle, and one on a raised platform aft. There are also two deck torpedo tubes.

Very special care was taken in the design and arrangement of the engine room to ensure that the ventilation should be as ample and efficient as possible, considering the inherent tendency of turbine spaces to be excessively hot. The after half of the aftermost funnel casing does duty as a huge up-cast immediately over the hottest part of the engine room—viz: the main steam pipes and their stop and regulating valves. In

addition, two ventilating fans are fitted to induce currents of air to keep moving in the required direction. The level of the starting-platform is kept low, and is so arranged that the engineer in charge has full control of the starting gear, the temperature at this point being quite normal. The extremely neat and compact arrangement of starting and maneuvering gear is worth noting. The large valves are all of Cockburn's patent flexible-disc type, and are easily operated by hand. The ahead and astern maneuvering valves for starboard and port respectively are each operated by a single hand wheel. The regulating valves to the cruising turbines are fitted on branches cast on the main regulating-valve box, and are placed underneath the maneuvering valves, one on each side.

The trials for this class of vessel are unusually exacting and exhaustive, and her builders, Messrs. Cammell, Laird & Co., Limited, are to be congratulated on the fact that their vessel is the first of her class to complete all the trials required by the Admiralty prior to fitting out for final completion ready for commissioning. For the high-speed trials it was decided to take the vessel to the Clyde, whither she proceeded at the end of September, and, on arrival there, a preliminary trial on the measured mile was carried out, and was succeeded by an official one-hour establishing trial at full speed, to determine whether the vessel and her machinery were in a sufficiently satisfactory state to make the full-speed six-hours' continuous-steaming trial, which, undoubtedly, is the most severe, as well as the most interesting, of the official trials that vessels of this type have to carry out.

The conditions under which this trial is made are of such a nature as to make it possibly disappointing to those who are in charge, owing to the fact that for the first three hours the vessel's speed is calculated from the number of revolutions obtained during that period, without the exact knowledge of what the actual speed may prove to be as the result of running the miles during the fourth hour; and from this cause a curious situation arose, as it was demonstrated on the fourth hour that the speed obtained was not equal to what had been antici-



H. M. DESTROYER "GHURKA."

pated by previous trial runs. In order, therefore, to obtain the speed of 33 knots over the complete period of six hours, special efforts had to be made to increase the revolutions, and consequently the speed, during the final two hours; and despite these efforts, the result, unfortunately, was that the average speed of 33 knots for the six hours was not quite realized, the actual speed being 32.977 knots, or a deficiency of 0.14 of a knot, or less than 300 yards in a total of 198 nautical miles, equivalent in time to 15 seconds.

Notwithstanding this very close result, the firm decided to make another trial, which was run a few days later, and the vessel was successful on that occasion in obtaining a speed of 33.1 knots, equivalent to 198.6 miles, during the six hours.

This mileage run during the six hours constitutes a record, inasmuch as no other vessel, we believe, has ever previously covered such a distance in the time named, although, no doubt, with the experience gained the result may be exceeded by some of the other firms who are building sister vessels.

We give below the actual speeds obtained in each individual hour in both of the above-mentioned trials:

	First trial. *Knots.*	Second trial. *Knots.*
Average speed for first hour	32.58	32.58
second hour	32.66	32.78
third hour	32.85	33.34
fourth hour	32.71	32.85
fifth hour	33.47	33.31
sixth hour	33.59	33.64
Mean speed	32.977	33.1

It must be borne in mind that at this extremely high speed the vessel is rapidly lightening as the trial proceeds, which accounts for the varying speed, although the speed during the fourth hour, during which the runs on the measured mile are made, is somewhat interfered with by the turnings at the end of each run when on the mile.

Shortly after the completion of the speed trial other subsidiary trials were carried out, and occasion was taken, during the vessel's return to Birkenhead from the Clyde, to run the

stipulated 24-hours' oil-fuel consumption trial at the economical speed. This trial, despite the fact that it was carried out in somewhat unfavorable weather, was entirely satisfactory, and dispelled a doubt which had existed in the minds of the builders as to what the result would be with this type of machinery, and the conditions under which the trial had to be made.

The contract radius of action of 1,500 knots was exceeded, the distance run being, on the stipulated consumption of oil, 1,580 knots, the other requirements being complied with in all respects.

The following are further interesting particulars of the vessel's performance:

Mean speed on the measured mile, 33.15 knots; highest speed on an individual run on the mile, 33.65 knots; consumption of oil for 24 hours, 27 tons; distance run per ton of fuel at the economical speed, 12.5 nautical miles.

Subsequent to the vessel's return to Birkenhead other important specified trials have been made, among which mention may be made of the gun-firing and torpedo trials, astern, circle, steering and anchor trials. On the occasion of the torpedo trials opportunity was taken to fire torpedoes whilst the vessel was steaming at full speed, and we understand the results that were obtained in this direction were eminently satisfactory and constituted a record as regards discharging at this high rate of speed.

Throughout the whole series of trials, including progressive trials, which were carried out by the builders, shaft horsepowers have been taken by "flash-light" torsion meters, supplied by Chadburns, Limited, of Liverpool. The results were checked and counter checked by turbine pressure and steam consumption in every possible way during the early trials, until, finally, the powers obtained were accepted with full confidence that the percentage error was less than obtains with ordinary steam-indicator diagrams. It has been of great interest to observe the variation in the distribution of power relatively through the three shafts, under different conditions of working the turbines, such as cruising alternately high and intermedi-

ate, and with slight overload. The shaft horsepower was easily read down to such low powers as 820, which was the power required at a speed of 13 knots, the lowest speed taken during the progressive trials. The facility of the system employed, including power diagrams for each shaft, enabled the shaft horsepower to be read off immediately on the completion of any interval of time during which the revolutions had been taken or at the end of each mile.

The vessel has proved herself throughout all the trials a thoroughly substantial sea-going vessel, absolutely free from vibration, and altogether one of the most interesting units that will be added to His Majesty's Fleet. She will now be pushed forward as rapidly as possible for completion and commission, and, we understand, will probably be delivered before the end of the present year.

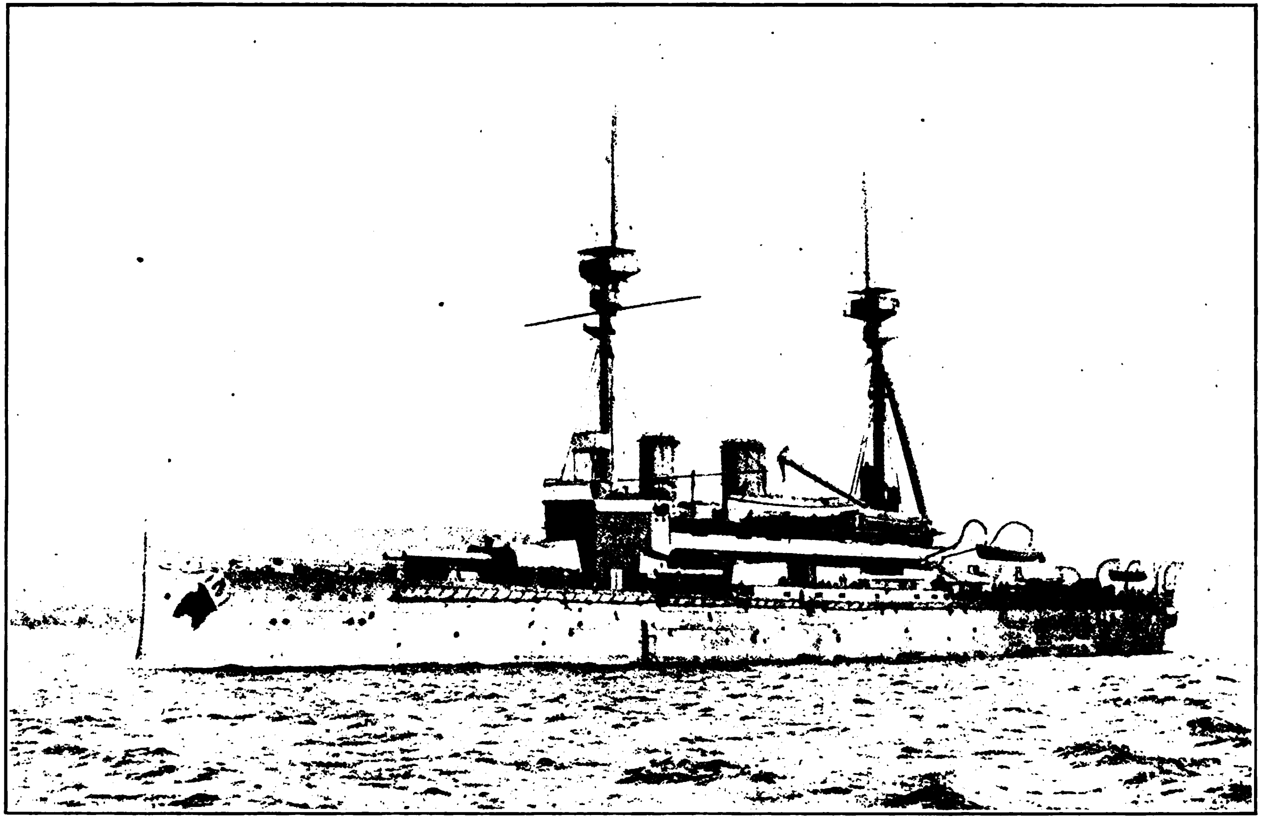
Messrs. Cammell, Laird & Co. have reason to be satisfied with the success of the *Cossack,* coming as it does in sequence to the various sizes and characteristics of destroyers since the date of their inception. This firm built five of the earliest of the 27-knot destroyers some thirteen years ago, and all of their vessels—the *Ferret, Lynx, Banshee, Contest* and *Dragon*—are still, we believe, in commission. These vessels were not equal in size, though termed destroyers, to the now up-to-date first-class torpedo boats.

Messrs. Cammell, Laird & Co.'s record through the various stages of the development of destroyers has been remarkable, they having built probably more of this class of vessel than any other firm in existence. No less than thirteen of the 30-knot destroyers have been built by them, the earliest of which was the *Quail,* launched in 1895, and the latest, the *Lively* and *Sprightly,* in 1900. The *Orwell,* one of the 30-knotters, it will be remembered, was cut in two in the Mediterranean, completely losing her fore end, and proved herself sufficiently seaworthy in that condition to reach Malta under her own steam, at which station her repairs were carried out.

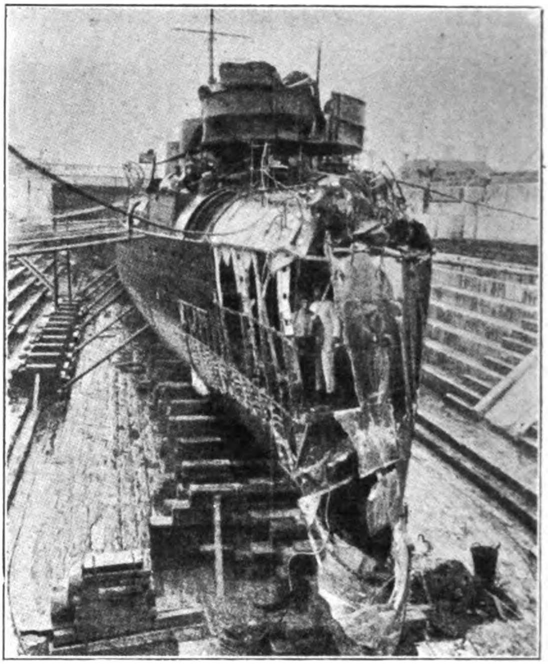
They have also constructed five destroyers of the River class,

a heavier and more seaworthy class of vessel, of 25½ knots speed, all of which have proved entirely satisfactory. They, however, have still to face a far more difficult problem than the *Cossack* in bringing to a successful issue the construction and trials of H.M.S. *Swift*, a vessel of about double the power of the *Cossack*, and a displacement of about 1,800 tons, and designed for a speed of 36 knots. This remarkable development of the torpedo-boat destroyer, starting from a displacement of 325 tons and attaining in the course of thirteen years to 1,800 tons, constitutes in itself a record of wonderful progress.—"Engineering."

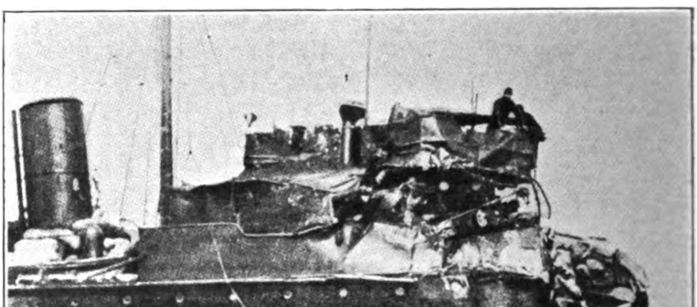
H. M. Battleship Agamemnon.—The new battleship *Agamemnon* attained on her speed trials a speed of 18.752 knots when the engines were developing 17,285 indicated horsepower, whereas the contract anticipated 18 knots when the machinery indicated 16,750 indicated horsepower. The *Agamemnon* is thus about equal in speed with the vessels of the *King Edward VII* class, and much superior to them in gunpower. As shown in the illustration, which is reproduced from a copyright photograph by Messrs. Maclure & Macdonald, of Glasgow, she mounts four 12-inch and ten 9.2-inch guns, instead of four 12-inch, four 9.2-inch, and ten 6-inch guns in the earlier vessels. There are many who prefer the *Agamemnon*, so far as armament is concerned, to the *Dreadnought* class, and two of the later ships built in Japan are equipped with two calibers, like the *Agamemnon* and *Lord Nelson*, while the French in their latest class are also adopting a combination of gun corresponding to that in the *Agamemnon*. This preference is due to the greater rapidity of fire of the lesser-caliber gun, but it must always be remembered that although these give a greater volume of shot per unit of time, their range for all of the guns is not so great. On the basis that the 9.2-inch gun may fire four rounds per minute as compared with two rounds of the weapon of 12-inch caliber, and that the 9.2-inch gun uses a projectile of 380 pounds, and the 12-inch gun a shot of 850 pounds, we have the following comparison of the broadside of the two ships:



H. M. BATTLESHIP "AGAMEMNON."



THE "KESTREL" AFTER COLLISION WITH THE "TEVIOT."



THE "QUAIL" AFTER COLLISION WITH THE "ATTENTIVE."

		Pounds.
12-inch	4 × 850 × 2 rounds per minute =	6,800
9.2-inch	5 × 380 × 4 rounds per minute =	7,600
Total per minute	=	14,400

and for the new *Dreadnought* class:

12-inch	8 × 850 × 2 rounds per minute =	13,600
Difference		800

With regard to direct bow or stern fire the superiority lies with the *Dreadnoughts:*

Dreadnoughts:		
12-inch	6 × 850 × 2 rounds per minute =	10,200
Agamemnon:		
12-inch	2 × 850 × 2 rounds per minute = 3,400	
9.2-inch	4 × 380 × 4 rounds per minute = 6,080	
		9,480
Difference		720

Of course the energy of the 9.2-inch guns is considerably less than that of the 12-inch; the former is capable of piercing 13½ inches of hard-steel plate at a range of 3,000 yards, against 19½ inches in the case of the 12-inch gun. The height of the center of the two forward 12-inch guns is 27 feet, and the two after 22 feet above the water line, and the average height of the centers of the 9.2-inch guns is 23 feet above the water line. The 12-pounder (18-cwt.) guns of the latest pattern are placed on the flying deck, a superstructure about 13 feet above the upper deck, extending from the forward to the after 12-inch gun barbette, and about 40 feet in width. The centers of these guns are therefore about 34 feet above the water line, and they are thus admirably situated in commanding positions for repelling torpedo-boat attack. The *Agamemnon* was built by Messrs. William Beardmore & Co. at their Naval Construction Works, Dalmuir, and the machinery was supplied by Messrs. Hawthorn, Leslie & Co., Limited, of Newcastle-on-Tyne.

Accidents to British Torpedo-Boat Destroyers.—There has been a regrettable series of accidents to torpedo-boat destroyers of His Majesty's Navy. With the exception, how-

ever, of the fire on board the destroyer *Spiteful,* the accidents were not attended with fatal results, although in one case some six or seven men were injured.

This latter accident occurred to the destroyer *Quail,* off the Isle of Wight. The *Quail,* one of the defending force, was engaged in assisting to repel an attack on Poole Harbour by the scouts *Attentive* and *Adventure.* The *Attentive,* traveling, it is reported, at about 20 knots, struck the *Quail* at right angles, cutting about 40 feet of the bows clean away. Being in action all water-tight compartments were closed at the time of collision, and the vessel remaining afloat was towed stern first to Portsmouth. The portion cut off by the *Attentive* sank, and several of the crew had rather narrow escapes, but no lives were lost.

Another accident occurred off Portland, to the destroyers *Teviot* and *Kestrel.* The vessels were both traveling at full speed at the time, with lights out, and the *Teviot* cut into the *Kestrel* for a length of about 30 feet. The damaged plating, etc., on the starboard side broke off and sank, while that on the port side was bent round, covering to some extent the gap on the starboard side. The *Teviot* did not suffer to the same extent, although her bows were buckled and damaged.

New Dreadnoughts.—Although the dockyard authorities at Portsmouth and Devonport are very reticent on the subject of the new *Dreadnoughts,* for which the building designs have already been received, there is reason to believe that the armament of these battleships will show no departure from the original *Dreadnought* specification, saving that the manner of disposition will be different, and that the anti-torpedo battery will consist of 4.7-inch guns instead of 12-pounders. It had been proposed to equip the new warships with 13.5-inch guns of an entirely new pattern, and several of these weapons have actually been built up. They weigh about 80 tons without mountings, and fire a projectile of 850 pounds. The idea had been to mount eight of these in each ship of the new group, instead of ten 12-inch guns, as in the original *Dreadnought.* But, so far as can be gathered, this project has now been definitely aban-

doned. The modified plan of distribution of armament in these latest battleships is the outcome of the discovery of imperfections in this way in the *Dreadnought.* At certain angles in the radius of her fire delivery several of the guns of this ship are masked by the turrets of other guns. Moreover, she could never discharge more than eight of her ten pieces in one broadside. In the new vessels the turrets are to be placed in a center line, so that the whole of the guns can be carried on either broadside. To obviate masking in either astern or ahead fire, the gun positions will be successively raised upon solid platforms, so as to give the rear weapons a clear range over the tops of the intervening turrets. The statement that it was proposed to place three guns in each turret is incorrect. Only two guns will be mounted in each of these citadels.

A 20,000-ton battleship is to be laid down immediately at the Devonport dockyard for the English Navy, being the fifth battleship of the *Dreadnought* class.

Temeraire, H.M.S., new battleship; 490 by 82 feet. There are now three ships of the *Dreadnought* class afloat—one in commission and two in course of completion. A fourth, the *Superb,* is building at Elswick by Sir W. G. Armstrong, Whitworth & Co. In design the *Temeraire* is like the *Dreadnought;* but, as in the case of the *Bellerophon,* recently launched at Portsmouth, some improvements will be introduced as the result of the experience gained during the trials of the *Dreadnought.* The vessel has a displacement of 18,400 tons. With turbine engines of 23,000 H.P. she will make a speed of 21 knots. As in the other ships of the class, the main armament will consist of ten 12-inch breech-loading guns, but it is understood that 4-inch guns will be mounted in place of the 12-pounders which the *Dreadnought* carries to repel torpedo attacks. The main defensive armor will be a belt of 12-inch hardened steel amidships, but there is likely to be a slight departure in the protective armament of other parts of the vessel. Among other improvements to be introduced is steering gear of a new description. The *Temeraire* was laid down on 1st January last, and is expected to be completed within two years

of that date. In the *Temeraire* and her two sister ships, the *Bellerophon* and *Superb,* a rearrangement of the 12-inch guns is introduced, the effect of which is to enable eight of these pieces to be fired astern as well as broadside. This has been done by raising the turret amidships on the quarter deck so as to permit the two guns which it contains to be fired over the after turret, which is also in the center line. The substitution of 4-inch guns for 12-pounders necessitates a rearrangement of the battery, and this, together with the alteration in respect of the big guns, has somewhat added to the displacement of the ship, which is 700 tons greater than that of the *Dreadnought.* Although the provision for the protection of the ships of this class against attacks by torpedoes or mines remains secret, there is reason to believe that in the last three vessels of the class not only have the armor arrangements been varied, but the cellular construction of the hull has been modified to provide a means of comparatively harmless exit for the gases set up by an under-water explosion. The whole of the machinery for this vessel is being constructed by Messrs. R. & W. Hawthorn, Leslie & Co., the turbine propelling machinery being of the Parsons type, driving four screws and giving 23,000 I.H.P., corresponding to a speed of vessel of 21 knots. —"The Steamship."

The new British 33-knot Torpedo-boat Destroyer Tartar. —The *Tartar*, one of the new class of 33-knot turbine destroyers, recently completed for the British Navy by Messrs. John I. Thornycroft & Co., Limited, of Southampton, ran a very successful contractors' preliminary trial in the Solent over the British Admiralty measured course on the 20th November, when the mean speed obtained on six runs was 33.122 knots, or 38.1 English land miles, without any special effort.

The vessel proved itself entirely free from vibration even at the highest speed, and in this feature stands alone among torpedo craft, though the introduction of the turbine has done a great deal to lessen the vibration usually associated with similar vessels using reciprocating engines.

The power is produced by turbines built by Messrs. Thorny-

croft, arranged on three shafts and developing approximately 14,000 I.H.P. together. The H.P. turbine on center shaft exhausts into the L.P. turbines on the wing shafts. Forward of the L.P. turbines is a H.P. and L.P. cruising turbine, each driving a shaft with one propeller. Steam is supplied by six water-tube boilers of the latest Thornycroft type, which use oil fuel exclusively, and work at 220 pounds per square inch. The fuel carried is sufficient for between 1,500 and 1,700 nautical miles.

The *Tartar* is 270 feet in length on water line, 26 feet in width and 17 feet 2 inches in depth. Her shape follows generally the lines of the earlier destroyers by the same firm, a prominent feature of which is the graceful turtle deck, which takes the place of the raised forecastle deck generally adopted for vessels of this size. The advantage of the former is that the water is thrown overboard instead of under the upper deck.

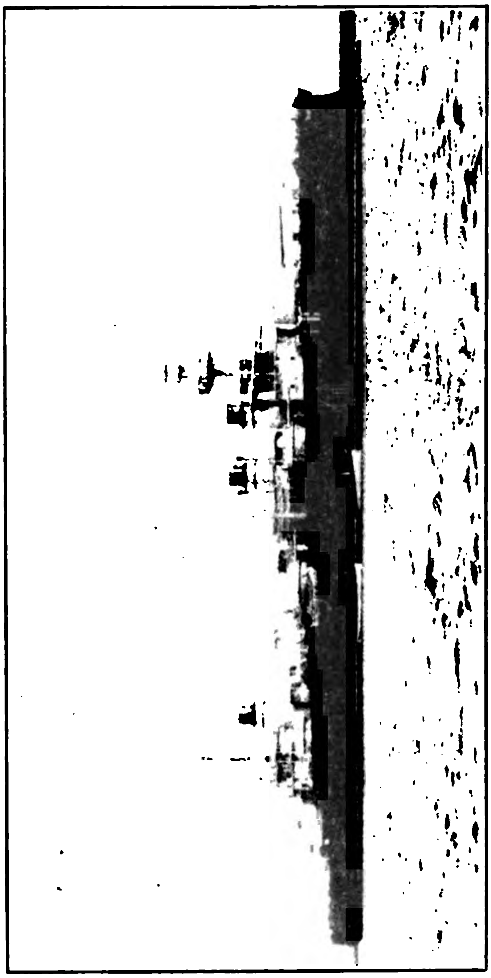
The armament consists of three 12-pounder guns and two torpedo tubes.

The vessel is of greater beam and freeboard than others of the same class, but nevertheless the speed mentioned above is the greatest yet obtained by any of the 33-knot boats on contractors' preliminary trials. From this fact it is safe to assume that by the time the vessel is tuned up ready for its official trials (which will take place very shortly, and consist in running six hours at a minimum of 33 knots per hour) there is every likelihood that still another speed record for torpedo craft will fall into the hands of Messrs. Thornycroft, who were the first to construct such vessels with a speed of more than 28 knots, and subsequently, for a considerable time, held the record for 30 knots and 31½ knots.

FRANCE.

Trials of French Battleships Democratie and Justice.—These two battleships, which have recently finished their official trials, are the third and fourth of the 1900 Program, being sisters of the *République* and *Patrie* and of the *Liberté*, and *Vérité*, which are nearly completed. The ships have a

load water-line length of 439 feet, an extreme beam of 79 feet 7 inches and a draught of 27 feet 5 inches. The displacement is 14,867 tons, with a block coefficient of 0.543. An armor belt, with a maximum width of 18 feet 5 inches, has



THE FRENCH BATTLESHIP "JUSTICE" AT ANCHOR.

an extreme thickness of 11 inches, tapering to 3 inches at the ends. There are two protective decks.

The battery, which is a powerful one, consists in the first two of the type of four 12-inch guns mounted in pairs in two

turrets, and eighteen 6.4-inch rapid-fire guns, of which twelve are in pairs in turrets and six are in casemates. The secondary battery includes twenty-seven 3-pounder and 1-pounder guns, and five torpedo tubes, two of which are submerged. A considerable difference exists in the battery of the last four ships of the type, through the substitution of ten 7.6-inch guns and eight 3.9-inch guns for the eighteen 6.4-inch guns on the first two vessels. This change makes for increased strength of broadside, particularly at long ranges. The 7.6-inch guns are located, six each in a single turret and four in casemates.

Each of the ships is propelled by three screws operated by triple-expansion four-cylinder engines in separate watertight compartments. The cylinder diameters are 35, 49, 55 and 55 inches, with a stroke of 41½ inches. Steam is supplied by twenty-four water-tube boilers, located in three boiler rooms, each with its own funnel. The total grate surface is 128 square meters (1,378 square feet), with a heating surface of about 46,000 square feet, making a ratio of 33.4 to 1. The boiler pressure is 251 pounds per square inch. The normal coal supply of 900 tons gives an estimated steaming radius at ten knots of 4,200 nautical miles. With the full coal supply of 1,850 tons the estimated radius is increased to 8,500 miles.

It must be pointed out that for the first time since the days of the old sailing fleet France possesses a homogeneous squadron of six vessels of high power, it having until recently been the French naval policy to construct contemporaneous vessels on the divergent designs of various naval architects, resulting in the acquisition of a fleet which was described by one of the French admirals as a "museum."

The first two ships of this type were compared in our August number with the battleship *Virginia*, of the United States Navy. The last four ships have a more powerful battery than the first two, but even with this advantage they could scarcely be expected to compete successfully with the ships of the *Virginia* class under ordinary conditions.

The results of the trials of the *Démocratie* and *Justice* are

given in the table. *Dêmocratie* is fitted with Belleville boilers, while the *Justice* has Niclausse generators. The poor propulsive efficiency at low speeds is more than offset by the magnificent results at 17 to 18 knots, while the figures for full speed are extremely good. It will be noted that in passing from the intermediate to the highest speed the index *n* in the expression $H_1 : H_2 :: V_1^n : V_2^n$ is very high, indicating a limit in highest propulsive efficiency at about 18 knots. This index, which is often assumed as 3, becomes 4.58 for the *Dêmocratie* and 5.98 for the *Justice*.

Six-hour trial at low power.	Contract.	*Démocratie.*	*Justice.*
Boilers at work	5	5	5
Grate surface, square feet	287	287	287
Indicated horsepower	2,500	2,584	2,541
Consumption of coal, in pounds:			
Per horsepower hour	...	1.234	...
Per square foot of grate	...	...	15.2
Mean speed in knots	...	7.5	7.8
Admiralty constant	...	98.8	112.8
Normal steaming radius	...	4,665	...
Twenty-four-hour consumption trial.			
Boilers at work	...	24	22
Grate surface, square feet	...	1,378	1,262
Indicated horsepower	10,500	11,472	11,520
Consumption of coal, in pounds:			
Per horsepower hour	...	1.484	1.528
Per square foot of grate	...	...	14.8
Average speed in knots	...	17.39	17.94
Admiralty constant	...	277.6	303.2
Normal steaming radius	...	2,027	2,023
Full-power three-hour trial.			
Boilers at work	24	24	24
Indicated horsepower	17,500	19,190	18,548
Consumption of coal, in pounds:			
Per horsepower hour	...	...	1.824
Per square foot of grate	24.3	24.3	23.7
Average speed in knots	18	19.44	19.43
Admiralty constant	201.6	231.6	239.1
Normal steaming radius	...	...	1,140

—"International Marine Engineering."

Latest French Submarine Boat.—The latest and largest type of French submarine boat, which is shortly to take its place in the Channel Defence Squadon, is illustrated in the engraving given in Fig. 3. She is named the *Emeraude,* and was ordered on the 24th October, 1903, and was laid down on December 1, 1904. She was launched on the 6th of August of last year, since when an extensive series of trials and experiments have been carried out, which are now completed. Her chief dimensions are: Length over all, 146 feet 6 inches; extreme breadth, 12 feet 10 inches; depth, 12 feet 1 inch. For ordinary navigation she is provided with two sets of internal-combustion engines, which together develop 600 horsepower. Each engine drives a propeller shaft. For submarine navigation a couple of motors are used. On trial a speed of 12 knots when three-quarters afloat, and 8 to 9 knots when half afloat, was attained. The *Emeraude* has a peculiar appearance, her hull being similar to the body of a large whale. The little craft, it is said, dives splendidly, but when afloat in rough weather she does not navigate so well as the "submersibles." For the officers and crew the accommodation is more comfortable than in the older type of boats, and, as will be seen from the engraving, the navigating bridge is large, and the visual radius for searching for an enemy is largely increased. Each of the three officers which are carried has his own room, and there are nineteen men in the crew. The armament is composed of six 18-inch torpedoes. The *Emeraude,* being the first of her class, has been subjected to very severe trials, but the five others, one of which has already been launched, will be completed in much less time. We are informed that the cost of building and equipment came to about £63,668.

In order that a comparison may be drawn between the *Emeraude* and what was hitherto the latest French submarine, we give an illustration in Fig. 2 of the "Y" type of boat. The Y has a length of 142 feet 2 inches, beam 9 feet 10 inches, and a draught of 7 feet 8 inches. It has a displacement of 213 tons and a horsepower of 250; her maximum speed above water being 11 knots, and when submerged 8 knots. She has two tor-

pedo tubes and carries a complement of fifteen officers and men. Fig. 1 shows the French submersible *Sirene,* which was laid down in 1900. She is 110 feet 6 inches in length, 11 feet 3 inches beam, and has a draught of 5 feet 2 inches. Her engines develop 217 horsepower and drive one propeller only. On trial she attained a speed of 12 knots. The armament comprises four torpedo tubes, and she is manned by two officers and eight men.—"The Engineer."

GERMANY.

German Torpedo Boat Construction.—The steady addition of torpedo craft of large size to the Imperial German Navy continues unchecked, and their value for war purposes is somewhat disguised by the official system of frequently describing them as torpedo boats, instead of classing them, as would be the case in England, as "destroyers." In general, their torpedo equipment is greater than that of British vessels of corresponding date, while their guns are fewer and less powerful; their hulls are of relatively heavier scantlings, and the vessels themselves have proved to be excellent sea boats, though slightly inferior in trial-trip speed to those of other nations.

No less than twenty-two boats of about 420 tons service displacement—*G 110* to *G 113* and *S 114* to *S 131*—were constructed by Messrs. Krupp, at the Germania Works, and by Schichau, at Elbing, between 1902 and 1905. Differing only slightly in detail, they are all about 205 feet long by 23 feet beam, and have bunker capacity for 100 tons of coal: at 15 knots the service radius of action is about 1,400 miles. Their trials were run at displacements varying from 380 to 415 tons, and speeds between 27 and 28 knots were obtained with about 6,000 to 6,500 indicated horsepower. *S 125* of the latter division, built by Schichau, was the first vessel in the German Navy fitted with Parsons' turbines. On her official trials she attained 26.7 knots at 440 tons, the increase in displacement being due to the greater amount of coal on board compared with her sister ships. The armament consists of three 4-pounder quick-firing guns, two machine guns, and three 450-mm. torpedo

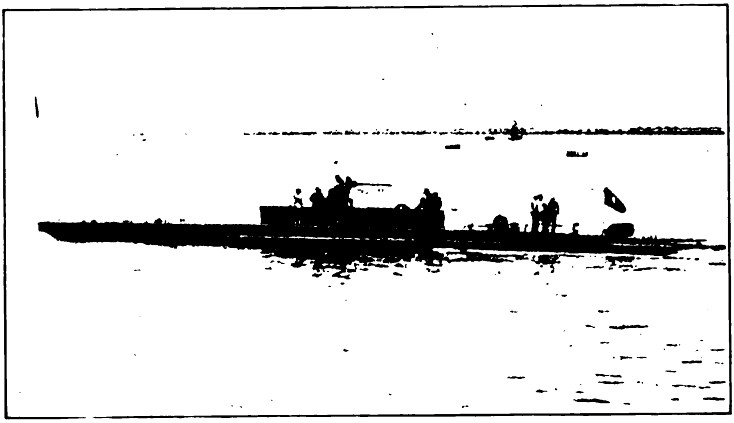


Fig. 1.—FRENCH SUBMERSIBLE "SIRENE."

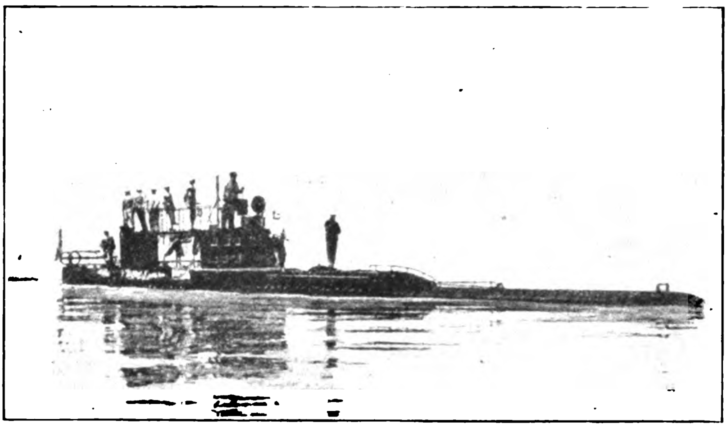


Fig. 2.—FRENCH SUBMARINE OF THE "Y" CLASS.

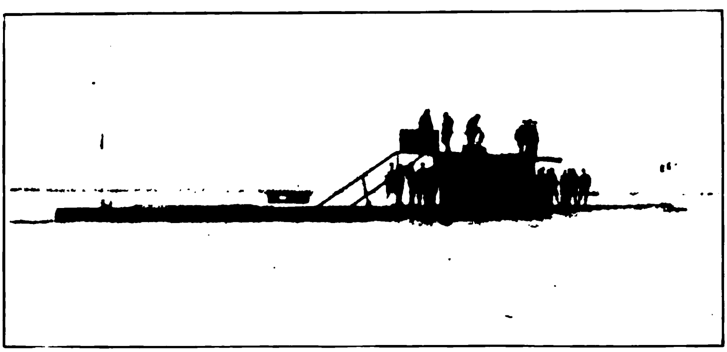
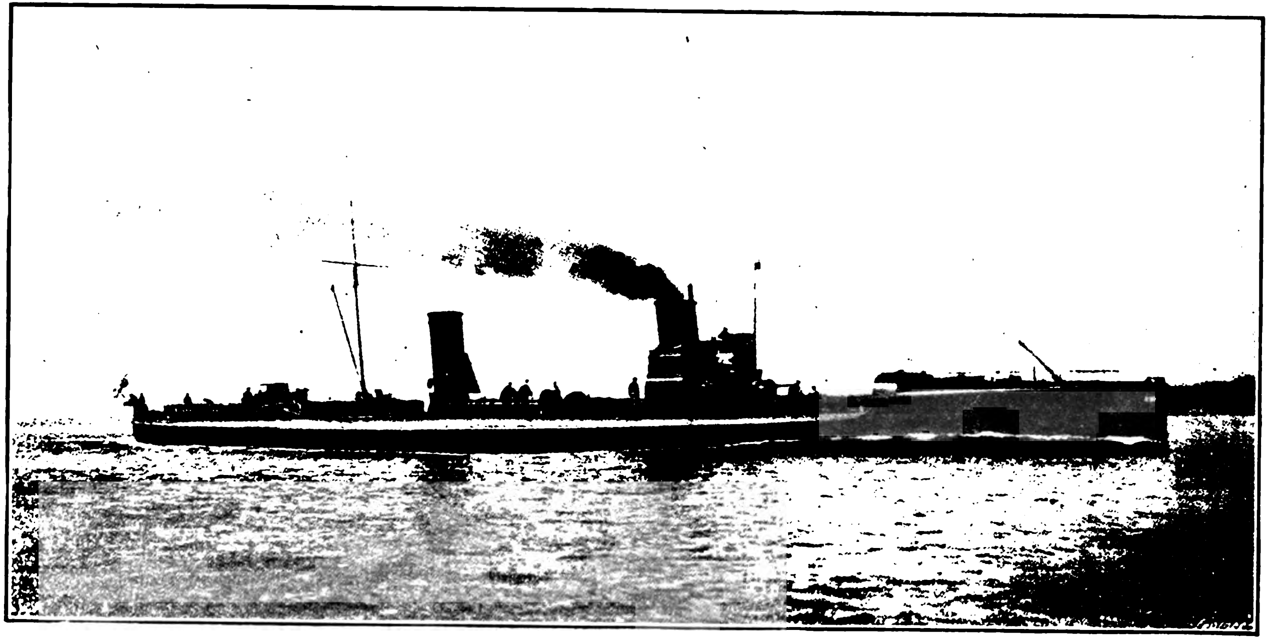


Fig. 3.—FRENCH SUBMARINE BOAT "EMERAUDE."



German Turbine-Propelled Destroyer "S 125."

tubes. Comparing these with British practice, we find that during the same period thirty-five "River" class destroyers of 570 tons were added to the Royal Navy, each carrying one 12-pounder and five 6-pounder quick-firing guns, but only two tubes. Their average full speed was barely 26.0 knots, but on their greater dimensions—225 feet in length by 23 feet 6 inches beam—they carry 125 tons of coal, and their radius at 15 knots is nearly 1,800 miles.

The next batch of German boats—*G 132* to *G 136*—was ordered in 1905, and consisted of 460-ton boats of 28 knots speed, carrying, as usual, three torpedo tubes, as well as four 5-pounder quick-firing guns and two machine guns. All these vessels were launched in 1906, and are now either in service or being completed at the Germania Works. They are almost identical with the previous destroyers, and, being built by only two firms, present fewer divergencies of appearance than is the case with the various British boats—Laird's and Palmer's, for example.

Since the commencement of 1907 several 30-knot boats, ordered in 1906, have been launched or have run their trials. *G 137* really belongs to this class, though she was ordered in December, 1905; the others consist of the Schichau division, *S 138* to *S 149*. These vessels all carry one 15-pounder, three 5-pounders, two machine guns, and three torpedo tubes. The designed displacement was 570 tons in the case of *G 137*, and 530 for the other boats; the indicated horsepower was to be 10,000. Four Schulz boilers are fitted, as against three in the earlier boats, the pressure being 230 pounds per square inch. In point of size they closely approach the dimensions of the "River" class, but are about 10 feet longer and 1 foot wider. About eight of the "S" class have run their trials, and, at a reduced displacement, have only attained 29.6 knots, though the horsepowers ran up to nearly 11,000. Considerable difficulty was involved in reaching even this speed, and numerous propellers were tried without success. The remainder of this batch is now approaching completion.

The latest destroyer division to be built for the German

Navy was ordered early this year from the Vulkan Company, which was entrusted with torpedo-boat work for the first time, and consists of twelve 580-ton boats *V 150* to *V 161*. Like *G 137* they are 235 feet long by 24.10 beam, and carry the same armament. As in most of the German destroyers, the engines are placed in two compartments, one abaft the other, and the coal bunkers, of 115 tons capacity, are extended at the sides of the engine room for purposes of protection. The designed speed is 30 knots, and in view of the performance of *G 137*, it will be interesting to see how these vessels compare with her. At present they are approaching the launching stage.

G 137 is fitted with marine turbines of the Parsons type, and on her recent official trials on the German Admiralty mile at Neukrug, attained a maximum speed of nearly 34 knots, and a mean speed for four hours of 33.1 knots, the vessel being run at full load. This performance makes her, for the time being, the fastest vessel afloat, and following on the failure of the sister ships of the *S 138* class to attain their designed speed, has created considerable interest in German naval circles where the turbine has hitherto been greatly disparaged. The main turbines consist of one high-pressure and two low-pressure cylinders of the usual type, arranged on three shafts, and revolve at about 850 revolutions per minute. Running astern, a speed of over 16 knots was obtained. The coal consumption averaged about 12 tons per hour at full speed.

No vessels corresponding to the large British destroyers of the *Cossack* type are as yet under construction, though it is possible that they may figure in the next Naval Estimates. For such speeds and displacements—33 knots and 780 tons—it is probable that oil fuel will be found essential, and this has not yet been applied to German torpedo craft.—"The Engineer."

The German Battleship Pommern has made a maximum of 20,348 horsepower, and 19.26 knots—a record for any German battleship so far, and well on the way to equal the French battleship *Liberté*. The coal consumption on the German ship was very high, while the *Liberté* accomplished almost a record in economy.

The speed of the Stettin is now stated to be 25.8. She is fitted with turbines of the Parsons type, and has proved about a knot better than her sisters, so far.

The German Navy League does not appear to be so satisfied with its Admiralty as is the British one. It is just conducting a strong agitation in favor of ships of 22,000 tons and 12-inch guns instead of 11-inch. Recent report has had it that the *Ersatz Sachsen* class are to carry two 12-inch per turret, instead of the original three 11-inch. The Navy League agitation suggests that the matter is still undetermined—a thing that should be useful powder to those of our contemporaries which expend so much energy in proving that the new German battleships do not yet exist.

Orders have recently been placed by the German naval authorities that indicate not only a remarkable change in their extremely antagonistic views of recent years on the subject of marine turbines, but also a genuine desire for definite information as to the relative values of different types of turbine. Cruiser "F," which is the first real continental reply to the British *Inflexible* class, has just been definitely ordered from Messrs. Blohm & Voss, of Hamburg, who have recently completed a splendid new machine shop for the construction of marine turbines. This vessel is to be completed within two years from now, and will prove a valuable addition to the German Navy. Her designed displacement is practically 20,000 tons, and her speed of 24½ knots will require about 44,000 indicated horsepower, which will be provided by turbines of the Parsons type. Exceptional care is being taken to keep the details of construction confidential; the main dimensions, however, appear to be as follows: Length on water line 560 feet, beam 85 feet, draught 27 feet.

Two small cruisers of 4,300 tons have also been ordered from the Vulkan yard at Stettin and from Messrs. Schichau, at Dantzig. The former will be fitted with a modified form of Curtis turbine supplied by the Allgemeine Elektrizitäts Gesellschaft, and the latter with turbines of the type introduced by Melms and Pfenniger. The latter turbine is a modified

Parsons, embracing many features of the Westinghouse combination system. The Krupp Germania works are also reported to have the order for a similar vessel which is to be fitted with Zoelly turbines. The speed in the case of these small cruisers is to be 25.5 knots.

GREECE.

Lonhi.—On July the 10th, in the presence of a large gathering, including official representatives of the Greek community in Great Britain, a torpedo-boat destroyer built for the Greek Government, was launched from the yards of Messrs. Yarrow & Co. (Limited) at Poplar, and received the name of *Lonhi*. The Greek Minister, who named the boat on behalf of the King of the Hellenes, was accompanied by Mme. and Mlle. Metaxas, and the staff of the Legation. After the religious ceremony by the Rev. Great Archimandrate Pagonis, the Greek Minister made a speech suitable to the occasion, and the launching took place amid the cheers of the company. The dimensions of the boat are: Length, 220 feet; breadth, 20 feet 6 inches; and depth, 12 feet 4 inches; I.H.P., 6,000; contract speed, 31 knots. The *Lonhi* is the third vessel of this class built by Messrs. Yarrow & Co. (Limited) for the Greek Government.

ITALY.

Much mystery still surrounds the Italian cruisers *Pisa* and *Amalfi*. These two ships of the *St. Georgio* class have been pushed on much faster than usually happens in Italy. The *Pisa* took the water at Leghorn recently, and the *Amalfi* is about to be launched at Genoa. The armament is four 10-inch, eight 8-inch; the horsepower 20,000. Belleville boilers have been ordered for them, not Blechyndens as originally reported. The *St. Georgio*, the first of the type, is apparently the only one with Blechynden boilers.

JAPAN.

Explosion on Japanese Battleship.—A terrible explosion occurred on the Japanese battleship *Kashima* on September 16th, while that vessel was at target practice off Kabutoshima in the

inland sea. The explosion, it is said, occurred inside a turret in which was a 10-inch starboard gun. The casualties first reported as the result of the explosion were: Killed, five officers and twenty-two men; severely wounded, two officers and six men; slightly wounded, two officers and six men. According to a later cable despatch from Mr. Dodge, Chargé d'Affaires of the American Embassy at Tokio, the explosion was more serious than at first reported. Mr. Dodge reports the loss of thirty-one lives instead of twenty-seven, and the injury of eleven men, and says that the disaster was "an explosion of shells." Naval Constructor Shinowara, Midshipmen Hayakiiwa and Nishimura were among those instantly killed. Lieutenants Arita, Fukuhara and Shigyo, and Midshipman Kitayama were said to be among those fatally wounded. After removal to the Kure Naval Hospital, Lieutenant Arita and eleven of the petty officers and men died. The Navy Department reports state that neither the gun nor the ship is seriously damaged. The *Kashima* is one of Japan's newest battleships, of 16,000 tons displacement, and was recently built in England. Among the officers and men killed were a number who had served with distinction in the Russian war. President Roosevelt at once directed the State Department to prepare a letter to Viscount Aoki, the Japanese Ambassador, expressing his sorrow at the news of the disaster on board the *Kashima.* Secretary Metcalf sent this despatch to the naval attaché at Tokio: "Convey to Minister of Marine the expression of the United States Navy's deep sympathy for the loss of life in the *Kashima* accident." The following official statement was submitted by Mr. Miyaoka to the Navy Department: "On September 16th, during target practice on board H.M.S. the *Kashima,* an explosion of charge occurred in the turret of the starboard aft 10-inch gun. The cause of the explosion is not clearly ascertainable owing to the fact that all officers and men on the spot have either been killed or severely wounded, but according to the statement of a man who left the turret just a moment before the explosion took place, it appears that when the accident occurred the gun in question had already been

twice fired. A new projectile was rammed and the charges were placed in position for the third fire, but the breech block had not been closed, when the said charge, catching fire from the backfire, set on fire the charge for the fourth projectile which was at the back of the gun. The projectile remained in the loading position. There was but little damage done either to the ship, the gun or the carriage. The number killed is: Officers and warrant officers, seven; petty officers and men, twenty-seven. The number of wounded, eight."—"A. & N. Journal."

SPAIN.

The Spanish Navy.—The reorganization of the Spanish Navy, which, according to official lists, only comprises one battleship, two armored cruisers, and six protected cruisers, besides five torpedo-boat destroyers, some gunboats and torpedo boats, will not only consist in the building of a number of up-to-date warships, but will also include the extension and reconstruction of the naval ports, and the construction of new naval fortifications. When the naval ports have been duly extended and enlarged, it is proposed to build the large ships in Ferrol, the smaller in Carthagena, and Cadiz is to have a large arsenal, more especially as far as artillery is concerned. Each of the three ports in question is to have its own floating dock. The ports of Vigo, Mahon, Ceuta, and Melilla will be fortified, but it is not proposed to make them arsenals. As far as the new fleet itself is concerned, the plan is understood to comprise six battleships, of 16,000 tons each, ten fast smaller cruisers, and twelve torpedo-boat destroyers. The ships first taken in hand will be three battleships and three torpedo-boat destroyers, and the armament will, it is understood, be specially adapted for coast defence. The reorganization of the fleet, and the works connected with it, will necessitate a State loan of 430,000,000 pesetas, which it is proposed to raise by an internal loan. The naval budget for next year will, according to the present plan, be increased from 35,000,000 pesetas to

50,000,000 pesetas. A considerable amount of trouble is anticipated in connection with the manning of the new fleet, and the English system of training is likely to be adopted.

SWEDEN.

The new Swedish battleships will be of 7,500 tons. Several are contemplated. Originally they were reported to be designed for four 12-inch guns, but now this has been changed. The present design is four 11-inch, four 7.6, eleven 4-inch, four one-pounders, and two 18-inch torpedo tubes. The belt will be 8 inches thick. The horsepower will be 17,500, and the speed 21 knots. Yarrow boilers will be fitted. Normal coal is only 450 tons, and the maximum capacity but 800 tons. The 7.6 guns seem rather superfluous, as they are neither one thing nor the other. An armament of six heavy guns would, we think, have been preferable.

The increased size of destroyers appears to be universal. Sweden is at present in the forefront in this direction, excluding the British *Swift*. The new Swedish destroyers are to be of 835 tons, 30 knot speed, and an armament of six 4-inch guns. The coal supply is very large.

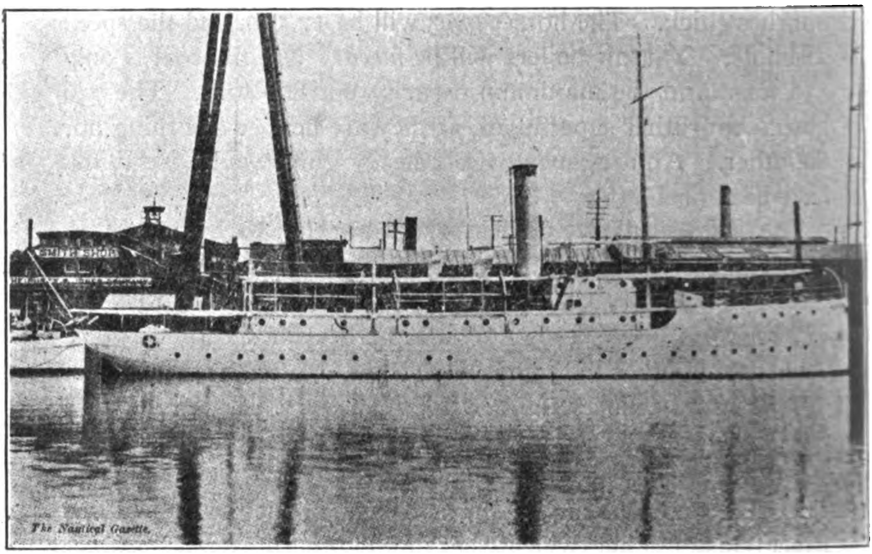
This quite eclipses the new German destroyers of the "V" type, of which twenty-four are ordered or being built—twelve for the 1906-07 program, and twelve for the current year. They are 520 tons only—that is to say, about the size of our River class. They will all be 30-knotters, however, and the armament one 24-pounder, three 6-pounders, and three tubes.

The armament of the Swedish destroyers, *Mode* type, is being altered from one 12-pounder, and five 6-pounders, to four 12-pounders.

UNITED STATES.

Revenue Cutter Pamlico.—The new revenue cutter *Pamlico,* built by the Pusey & Jones Co., Wilmington, Del., for the Revenue Cutter Service, is for the North Carolina sounds. She was recently placed in commission at Baltimore, with First Lieutenant H. G. Fisher in command, and Second Lieutenant J. L. Maher as executive officer.

The *Pamlico* is a shoal-water, single-mast, twin-screw steamer, and cost in the neighborhood of $175,000. The principal dimensions of the steel hull are as follows: Length over all, 158 feet; length between perpendiculars, 148 feet; depth from base line at side, amidships, 10 feet; spring of beams in 30 feet, 7½ inches; breadth of beam, molded, 30 feet; draught of vessel with 25 tons of coal and 2,400 gallons of fresh water, stores and provisions, ready for sea, 5 feet; displacement at 5-foot draught, 408 tons.



NEW REVENUE CUTTER "PAMLICO," BUILT BY THE PUSEY & JONES CO., WILMINGT

The hull is divided into six compartments by five watertight bulkheads. On the top gallant forecastle is located the steam windlass engine, anchor davits, and a 6-pounder rapid-fire gun. From this deck a hatchway and ladder afford access to the main deck. The top gallant forecastle deck extends aft to the top of the deck house, thus forming a continuous deck, affording protection to the main deck forward.

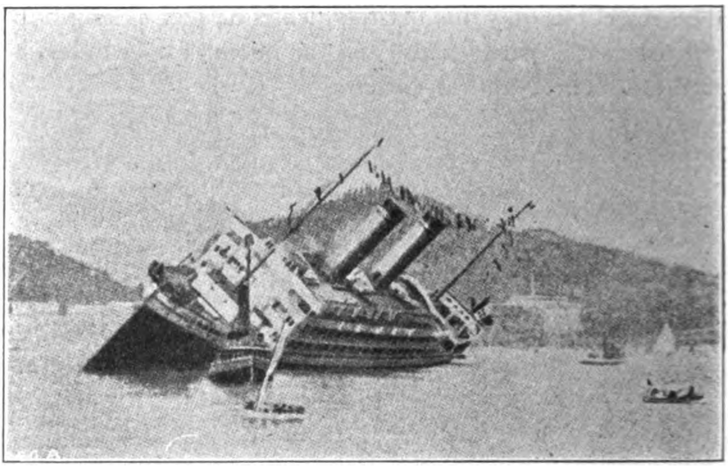
There are two triple-expansion propelling engines of the vertical, inverted cylinder, direct-acting, type, each having one

high-pressure cylinder 8 inches, one medium-pressure cylinder, 13¼ inches, and one low-pressure cylinder, 22 inches in diameter, the stroke of all pistons being 18 inches. There will be one Sirocco fan blower driven by steam turbine, located in the boiler compartment. It is designed to run with a steam pressure of 100 to 120 pounds, and will be connected with ducts to the ashpits, and to the ventilating system of the vessel. Steam is supplied from a boiler of the marine water-tube type, 9 feet 3 inches in length, 11 feet 9 inches in width and 11 feet in height, working at a pressure of 200 pounds to the square inch. The boiler for the donkey engine is of the Scotch type, with a working pressure of 200 pounds to the square inch.

The vessel has only one mast, which is 64 feet in length over all, fitted with yard arms and a boom, for wireless telegraphy.—"The Nautical Gazette."

MERCHANT SHIPS.

The Disaster at Riva Trigoso, Italy.—A feeling akin to consternation has been spread through Italian shipping circles owing to the mishap in launching the biggest passenger steamer yet constructed in Italy. The building yard is situated at Riva Trigoso, a small place about halfway between Genoa and Spe-

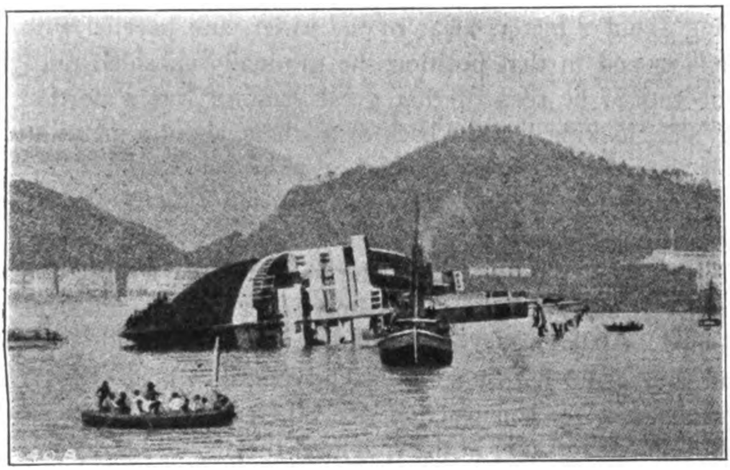


zia. It was started in 1889, and has gradually increased until now it has an area of 40,000 square meters, and three building slips of masonry, capable of taking vessels of 150 meters (492 feet) in length. These slips are served by seven radial electric cranes capable of lifting a weight of 2 tons to a height of 25 meters, with a radius of 20 meters, which were described in one of the papers read at the last session of the Institution of Naval Architects.

The size of the vessels constructed has gone on increasing, until the last to be laid down were the biggest yet constructed in Italy. These two vessels, to be known as the *Principessa*

Jolanda and *Principessa Mafalda,* were for the South American service of the Societa Lloyd Italiano. They are 149 meters long (489 feet), with a beam of 17 meters (56 feet), and were to have a displacement of 12,000 tons. They have twin-screw quadruple-expansion engines of 10,000 horsepower, which were to give the vessel a speed of 18½ knots. The first of these two vessels—the *Principessa Jolanda*—was finished complete, with engines and boilers, and was to be launched on September 22d, whilst the second vessel is well advanced, all her framing being completed.

The day fixed for the launch was a typical Italian day—



bright sunshine, a cloudless sky, and delightful temperature. The trains from both directions of the Riviera took hundreds of eager spectators to the place, and many steamers, large and small, took other visitors down from Genoa, and the sea being quite smooth, these were largely patronized.

Shortly after midday the naming took place in due form, and the bottle of sparking asti, decorated with flowers and the Italian colors, was broken in the traditional manner. Twenty minutes afterwards the signal was given that all was clear, and the vessel gradually began to move, quickly increasing her rate of speed, and amidst the cheering of thousands, the hoarse

tones of the whistles, and the shrieking of syrens, the launch was quickly and triumphantly completed.

But no sooner was the vessel fairly afloat than she was seen to keel over in an alarming manner; the cheering ceased in an instant, and a dead silence followed; the effect of this and the huge mass of the vessel slowly going over was so horrifying to the spectators that they started to flee from the spot.

The tugs had at once got hold of the vessel, and she was pulled round parallel to the shore. The inclination was to port, and by this time the water had reached the port-holes of the main deck, some of which had been left open, and the cabins quickly filled. She was soon on her beam ends, her funnels being about 2 meters clear of the water, and parallel with its surface, and in that position she gradually subsided until all that was to be seen of that great steamer was a portion of her side, looking like the back of a whale, about a meter and a half above the surface at its highest point.

The effect on the spectators was intense. The shipyard is so situated in the corner of the bay that the vessel seemed to dominate everything. That she should have entirely disappeared in such a fashion was horrifying to the onlookers; it seemed as if some terrible nightmare had got hold of them, and they could hardly speak. The officials who had been responsible for the construction stood there as if turned to stone, and remained gazing at the spot as if they could hardly believe their eyes, whilst the workmen who had built her, and who, with their families, had almost cheered themselves hoarse a few minutes ago, now were weeping and hugging one another in a state bordering on delirium.

Bad as the disaster was, it was not rendered still worse by loss of life, for the disappearance was so gradual that all on board were got off before the hull went under.—"Engineering."

A Motor-Driven Liner.—There is now under way in England an experiment which, if successful, will mark a new step in marine propulsion and achieve results by which the *Lusitania's* speed record will be put in the shade. The keynote of

the idea is the application of electricity to turbines, and a well-known firm of engineers is equipping a vessel with an apparatus designed to make the test both practical and complete.

It must be remembered that the steam turbine is most efficient when running at high speed, while a ship's propeller, on the other hand, will not work efficiently at the highest speed. If the speed be increased beyond a certain point, far below the most efficient speed of the turbine, the blades of the propeller simply churn the water instead of driving the ship. It is impossible to gear down from a turbine to a propeller shaft, for the horsepower of marine turbines is too great for any practicable form of gearing. Consequently the turbine has to be run slowly, and an inevitable loss of efficiency in this direction is put up with.

The speed of the *Lusitania's* turbines is only 180 revolutions per minute, and to adapt them to these conditions they are large in diameter and have blades of great sectional area. This means that there must be sparse clearances, and these in turn mean that steam entering the turbines at high pressure finds its way toward the condenser without giving out the whole of its heat and energy. If brought to perfection electrical transmission will form a link between the swift turbine and the slow propeller.

The plan upon which the firm of engineers which is now preparing to make the practical test spoken of is not that the turbine should be coupled directly to the propeller shaft, as is now done, but should drive high-speed electrical generators and supply current to electrical motors for driving the propellers. A qualified engineer further explains the idea as follows:

"For fast passenger boats the arrangement will resemble that of a modern electricity-supply station with the many units of the plant feeding the common system of mains. All the turbo-generators will supply the common bus-bars on the switchboard, so that it will be possible to feed any motor from any generator. The system of supply adopted will be either continuous or alternating, preferably the latter, as high pressures

can be used and commutation troubles avoided. Some alteration in the disposal of the machinery would be necessary, but on the whole there would be a gain of space. But more important than any consideration of space, the electrical system possesses the advantage that the motors can be reversed almost immediately."

A future *Lusitania* may be driven by turbo-generators of 100,000 H.P. at a speed of 30 knots. Such a vessel would have six turbo-generators of 20,000 H.P. each, one of which would be in reserve. Each of her four propellers and the shafts would be provided with six motors of 5,000 H.P., five of which would do the work while the other would be a standby, running light but ready on the pressure of a button on the bridge to take up its share of duty.

For the bridge electrical transmission will mean a revolution, the navigating officer will no longer have to signal his orders for the maneuvering of the ship to the engine room. He will have beside him a keyboard of push buttons by which he himself will control every movement of the ship instead of ordering the engineers. To go astern, for example, he will push a button which will reverse the motors, and so with every variation of speed and direction.

Such an accident as occurred to the *Deutschland* in Dover Harbor would be impossible. The eye that sees the danger and the hand that prevents disaster will be controlled by one brain, and the navigating officer on the bridge, conscious of imminent peril, will not have to transmit mechanically his orders to the unseen engine room below, where their immediate performance, on which the vessel's safety depends, may be hindered by slow comprehension or an accident of some other nature.

In the electrically-propelled ship the eye that sees and the brain that understands will alike control the propeller and the helm.—"The Marine Review."

MANUFACTURERS' NOTES.

WATERPROOF AND STEAMPROOF LEATHER BELTING.

Waterproof and steamproof leather belting was exhibited, for the first time in history, at the Jamestown Exposition. Charles A. Schieren & Co. have received a gold medal, the highest award, from the exposition for their Duxbak Leather Belting on the distinct claim that it is a waterproof and steamproof belting. Charles A. Schieren & Co. have been perfecting their waterproof belting for several years but never until about a year or two ago was it advertised under a trade-mark name and with the distinct claim that it was waterproof and steamproof. They claim that a distinct and important step forward has been taken in the making of leather belting in their Duxbak process.

The exhibit of the Schieren Company at Jamestown has attracted widespread attention throughout the months of the Exposition. It is a beautifully-painted glass picture representing a lake, out of which flows a real waterfall upon a roll of Schieren's "Duxbak" belting. The exhibit is designed on the lines of an interior view of a camera. From the front of the booth black cloth is draped artistically on the two sides and top to the rear of the booth, throwing the brilliantly-lighted picture out into bold relief. The size and magnificence of this display can best be realized when it is known that the beautiful big roll of "Duxbak" belting is eight inches in width and forty-two inches in height. The rear of the plate of glass and the frame are lit up with 148 electric lights. Appearing and disappearing signs occupy either side of the display. On one side the "Duxbak" trademark appears, followed on the other side by the words, "Chas. A. Schieren & Co., New York."

The water falls from a fissure in the glass so naturally that it appears a natural waterfall from the lake itself. It

drenches the entire surface of the belt coil, and the pond formed by the overflow covers the bottom part of the coil to the depth of eight inches. Notwithstanding the fact that the belt is soaked all day long in this depth of water in the tank, it is perfectly dry and flexible when the water is turned off at night and the tank drained. In fact, the belting is so firm and solid and has so little affinity for water that it hardly appears at all wet the instant the falls are turned off. In order to further demonstrate the absolutely waterproof character of the "Duxbak" belting the end of the roll is uncoiled and is erected under the falls so that the sides of the roll are thoroughly drenched. The water hits the belt just at the end of the lap, and plays continuously all day long right in the cement—a test to which no belt has previously been submitted. This particular coil of belt has already remained under the water $5\frac{1}{2}$ months and it is the intention of Chas. A. Schieren & Co. to leave it there during the remainder of the Exposition, having confidence that it will be as good as new at the end of that time. This is certainly convincing testimony as to the waterproof character of "Duxbak" belting. Visitors are greatly impressed by the substantial and honest character of this test, and everyone declares that in the line of belting the "Duxbak" has proved the best thing they have seen so far.

This is, in fact, the severest test to which a belt can be subjected. When it merely revolves in water the tendency is to throw the water off the surface of the belt, but with the waterfall under high pressure, as in Schieren's demonstration, the water strives to enter every fiber of the leather, but without avail, since Schieren's waterproof dressing sheds it like a duck's back, and the waterproof cement makes the laps impervious as well.

The balance of this electric illumination is also very attractive. The water in the tank is bordered by a semicircle of palms and ferns. On both sides of the booth, extending from the tank to the front of the booth, are arranged symmetrically fourteen rolls of the Schieren belting.

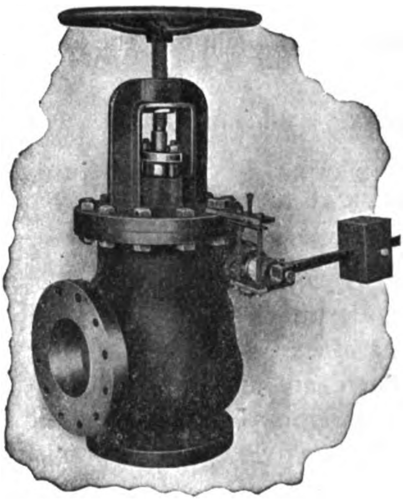
The exhibit is indeed unique and clever in a number of ways, and might be taken as a good object lesson by those advertising men who say Expositions cannot be made to pay. It has movement and life, the first of all requisites to draw a crowd. It has artistic merit; it has color; it makes prominent the trademark of "Duxbak" belting, all most important considerations. Best of all, however, it convinces the crowd and makes a lasting impression by its clear argument when the crowd has been attracted. The vast majority of such exhibits most assuredly do not pay in traceable returns for the money invested—but the reason for this is not the poorness of the Exposition at which they are shown, or any other weakness in Exposition advertising as such, but the lack of merit of the exhibit itself.

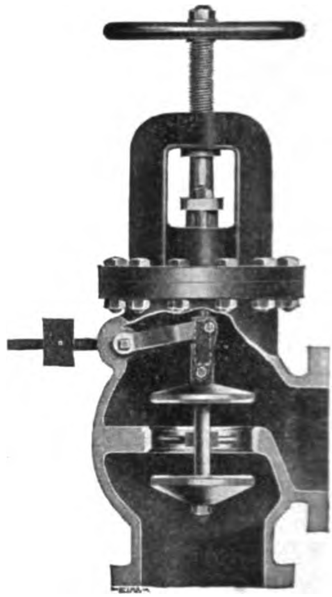
ADVANTAGES OF AN AUTOMATIC CUT-OFF VALVE AS SHOWN BY TESTS AND PRACTICAL EXPERIENCE.

By A. Eugene Michel.

Such great damage is done by the occasional bursting of steam mains or the failure of boiler tubes, where there is high steam pressure, that some kind of excess-flow safety valve is essential for every boiler. There are many devices on the market which will prevent steam from the mains from flowing back into the boiler, but these valves make no provision against accident to the steam piping. The escape of steam would continue until the boiler were emptied. The valve which we are about to describe not only acts as a "non-return" valve, but it also shuts off the boilers with equal surety and protection if the main should be broken, as from water hammer or from the breaking of an elbow. As a cut-off valve it acts automatically, but by the turning of a hand wheel it may further be used as an ordinary stop valve. As steam is raised in the boiler the valve opens it to the main when the boiler pressure is equal to the pressure in the line, thus avoiding all accidents from carelessly opening valves, while there is considerable difference of pressure.

This valve works instantly either way and does not depend upon difference in pressure for its action, but upon the actual flow of steam through the valve. If a tube in one of the boilers should give way it shuts down that boiler only, and allows all the other boilers in the battery to go on supplying steam as usual. With an ordinary stop valve the fireman is frequently scalded in trying to shut off the main or is driven from the fireroom without being able to do anything to save the entire plant from shutdown. This cut-off valve will, un-

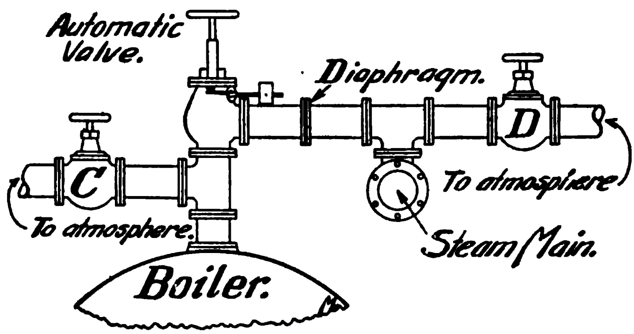




less intentionally opened, stay closed until the pressure is raised again. It is, therefore, perfectly safe for a man to go inside to repair damages as soon as the injured boiler has cooled off sufficiently.

If, on the other hand, a steam header bursts, or a joint breaks or a cylinder head blows off the engine, the cut-off valves on all the boilers close immediately before the room has been filled with steam, and repairs can proceed at once. The operation of this valve may easily be seen from the accompanying cross-sectional view. It is installed so that the

lower valve disc is toward the boiler and the hand wheel on top. Normally, when the boiler is not working the upper valve rests upon the seat and prevents steam from the main from entering the boiler. When the steam pressure in the boiler is raised to slightly exceed that in the main, the valve lifts and steam flows from the boiler into the main. The valve is very nearly counterbalanced by a weight on an external arm. The leverage of this weight may be adjusted to different rates of steam flow for boiler output. The valve is operated, not by pressure, but by the actual flow of steam through it. The normal flow of steam into the main raises the discs to mid position, as shown. The valve remains in that position as long as steam is being drawn from the boiler



at the normal rate, but in case of a break on line side of the valve the excessive rush of steam would carry the lower valve up against the seat, shutting off the boiler. Of course, when the flow reverses the upper valve drops instantly to its seat and shuts off the steam. The rate of flow at which the boiler would be shut off is determined by the weight above mentioned, and by the distance between the two valve faces. This is adjusted to correspond to the greatest overload at which it is desired to operate the boiler, say a rate over twice the normal rated capacity, or when the water begins to raise in the gauge glass.

A fork-and-link arrangement connects the valve discs and balancing lever positively, so that the position and condition

of the valve may be determined by a glance at the balancing lever.

The balancing lever is provided with light springs which prevent the chattering or closing of the valve under ordinary conditions. The springs are not strong enough to prevent closing in case of accident, but will prevent closing in case of a momentary rush of steam. These springs can be adjusted to meet different quantities of flow of steam.

The following tests made on this valve in the power house of the Anaconda Mines, Anaconda, Mont., illustrate how it acts in case of accident. The accompanying illustration shows the arrangement for the test. The automatic valve was placed between the boiler and the main. On the boiler side was a quick-opening valve "C" communicating with the atmosphere, and on the main side was another quick-opening valve "D," also opening to the atmosphere. The boiler was a 300-H.P. Stirling water tube, carrying a pressure of 150 pounds. When this pressure was reached the valve "D" was opened wide enough to allow steam to escape from the boiler to correspond with different rates of driving. To determine how the valve would act if the main should burst, the valve "D" was thrown suddenly wide open. The water rose so rapidly in the glass that if it had been allowed to continue, it might have resulted in the destruction of the boiler. Before the water had gone up two inches, however, the automatic valve closed tight. A test was then made to see if the automatic valve would close if the break opened slowly. A steel diaphragm was placed in the pipe as shown; in this diaphragm was an opening 1⅝ inches in diameter, and when the valve "D" was opened suddenly the automatic valve closed promptly. The diaphragm was then removed and another substituted with an orifice 1½ inches diameter. When the valve "D" was opened again the automatic valve did not close, showing that its point of closing was between these rates of delivery. In this case it was set to close at a flow corresponding to 600 boiler H.P.

After making these tests the boiler was again placed in

service and a test was made to see what would happen if a boiler tube should break or some part of the boiler give way. This was done by suddenly opening valve "C," upon which the automatic valve closed promptly, shutting off the main. Fifty-four tests like these were made and the valve acted perfectly each time. The tests were so satisfactory that sixteen of these valves were installed in the plant.

The manufacturers frequently receive reports of accidents where the valve more than pays for itself in one emergency. One of these valves had hardly been installed in a plant recently when a blind end blew off the steam header. The engineer had forgotten to put in all the bolts, but the automatic valve acted before further damage could be done.

Bursting steam pipes frequently scald their victims to death where there is no automatic means of shutting off the flow, but the danger in refrigerating plants is even greater. Just what ammonia will do if the flow is not stopped by a cut-off valve may be judged from accidents which have happened at Armour's Refrigerating Plant in Chicago. A cylinder head blew off an ice machine last January while twenty men were in the room. Three were killed, sixteen overcome by the fumes before they could get out, and only one escaped. In May a two-inch ammonia pipe burst, and the fumes killed six men, 200 head of cattle and a thousand sheep. The ammonia fumes were so strong that rescuers could not enter for four hours. A cut-off valve in the line would have checked the fumes in both instances before enough would have escaped to be serious.

The Lagonda automatic cut-off valve is made by the Lagonda Manufacturing Company, Springfield, Ohio, who will be pleased to send further descriptive literature upon request.

THE NEW PLANT OF THE LAGONDA MANUFACTURING COMPANY.

Twenty years ago, when water-tube boilers first began to be introduced, a man named Weinland invented a machine for cleaning scale from out the tubes. It was of the type now

known as the "turbine." When these cleaners were put on the market the business was carried on in a 10-foot by 12-foot room, the cleaners being made outside. The venture proved successful, and the cleaners came into favor so rapidly that machinery for their manufacture was purchased and installed in a room 24 feet by 24 feet. The company then began to take contracts to clean boilers. In that way the merits of the cleaners were demonstrated to possible customers, and those who did not care to bother with the cleaning themselves were assured of having the work done quickly by competent men. Again the quarters became too small. This time the plant was moved into a 28-foot by 70-foot two-story iron-frame building (built by the present company), and every effort was made to build the most and the best cleaners used in this country.

As water-tube boiler plants became larger and larger the scale problem assumed more importance, and the time that boilers had to be laid off for cleaning represented a heavy loss of revenue, so the Lagonda Manufacturing Company, the concern referred to above, sought to devise some machine that would permit the application of greater power to the scale-cutting device than was possible with the turbine cleaner. The result was their Weinland Mechanical Boiler-Tube Cleaner, which can be propelled through the tube by any power, from a 4-H.P. water motor to a 25-H.P. steam engine or electric motor.

Where scale is exceptionally hard or thick, or where there are so many boilers that cleaning becomes a heavy item of expense, or where the water for driving turbine cleaners would cost considerable, the power-driven mechanical cleaner is by far the most economical device that can be used. It is also much cheaper to maintain, as its parts are almost indestructible and are cheaply replaced. The Weinland Mechanical Cleaner is used almost exclusively by the Lagonda Company in its boiler-cleaning contract work.

The Lagonda Manufacturing Company has also added other specialties and, with the increasing demand for the Weinland

cleaners, has found greater manufacturing facilities necessary. A new plant has therefore just been built, consisting of a two-story 50-foot by 200-foot building, located on two acres of ground, which will be available for further growth.

The new building, although not first in size among modern factories, is a model in the character of its construction and equipment, and it is undoubtedly the largest plant in the world building boiler-tube cleaners. The walls and floor are of artificial stone, and both floors are well heated, lighted and ventilated. It has modern wash rooms, lockers and toilets. Every machine tool is driven by an individual motor receiving power from a gas-engine-driven generating plant. The absence of line shafts and belts and the large windows give plenty of light and make the shop an ideal place for workmen to do their best.

All the lathes and automatic machines are made with special attachments for turning out the peculiar bearings, water wheels, arms and other small parts accurately and in great quantities. The equipment includes furnaces for heating oil and lead baths for tempering an elaborate testing bay, for trying out each finished cleaner before allowing it to leave the factory. This testing department contains tanks, pumps and gauges of proper size to subject every cleaner to actual working conditions.

The cut-off valve department, is devoted to the manufacture of a device placed in the steam connections of boilers for automatically stopping the passage of steam in either direction in case a steam main bursts or a boiler tube gives way. The sales of this valve seem to indicate that engineers are quick to appreciate the protection which it affords. Many "non-return" valves have previously been offered to steam users, but this is the first practical one which works equally well in both directions and which contains no pistons or other sliding parts which could stick or bind.

FREE ENGINEERING LIBRARY TO OPEN EVENINGS.

On and after Wednesday, November 6, 1907, the reference libraries of the American Institute of Electrical Engineers, The American Society of Mechanical Engineers and The American Institute of Mining Engineers, 29 West 39th Street, New York, will be open evenings until nine o'clock on all week days except public holidays.

These Libraries, constituting practically one library of engineering, situated near the New York Library, in the new headquarters of the Engineering Societies, are available to members of the above Societies, engineers, and the public generally, subject to proper regulations. Strangers are requested to bring letters of introduction from members or to secure cards from the Secretaries of the respective Societies.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The fifty-fourth annual meeting of the American Society of Mechanical Engineers will be held in the Engineering Societies' Building at 29 West 39th Street, New York, December 3–6, 1907.

Symposiums on foundry practice, giving the experiences of prominent men in that work, have been arranged. The specific heat of superheated steam will be taken up, a very important and exhaustive work by a Professor of Engineering at Cornell, will be presented. The utilization of low-grade fuels in gas producers, combustion control in gas engines, tests of producer-gas engines, etc., will be given a session. Other live topics, such as industrial education, power transmission by friction driving, cylinder port velocities, etc., will be discussed.

All of these subjects have been treated by prominent engineers of Europe and America, professors of our universities, and men eminent in the particular work of which they write.

The Committee have on hand an interesting excursion for Wednesday afternoon and an address in the evening which will be especially enjoyable.

BOOKS RECEIVED.

Electrical Installations of the United States Navy. A manual of the latest approved material, including its use, inspection, care and management, and method of installation on board ship. By Commander Burns T. Walling, U. S. Navy, and Julius Martin, E. E., Master Electrician of the Equipment Department, Navy Yard, New York. Published by the U. S. Naval Institute, Naval Academy, Annapolis, Maryland. Price $6.00, post paid.

This very excellent work, consisting of 648 pages, gives a most comprehensive description of the electrical apparatus in the Navy, together with full information as to the inspection, care and handling.

Its value to those who have charge of the electrical machinery on board our naval vessels cannot be overestimated. The student will find it equally valuable. The book is divided into fifteen chapters, under the following heads:

Chapter.

MODERN MARINE ENGINES WITH THEIR AUXILIARIES AND APPURTENANCES, LAUNCH MOTORS AND STEAM TURBINES. For educational and practical purposes. By ROSENTHAL, MUELLER and BAYER, Instructors of Engineering at the School of Navigation, Hamburg, Germany. K. M. MECKLENBURG, Berlin, Publishers.

The value of this interesting work is greatly enhanced by a volume of 53 plates, lithographed in colors, containing over 1,200 illustrations of machines. Its size, 8 inches by 11 inches, is very convenient and handy for reference since there are no folders.

The work throughout contains only practical material. The drawings are neatly executed, without unnecessary complications by figures, and the tinting of the sections in colors denoting the materials, adds not only to the appearance but also to the clearness of reading. This method is certainly much to be preferred to the usual cheap and clumsy methods of cross hatching in one color, and is well worth the additional expense.

Originally intended only as material for the course of instruction and with the intent to assist the student in the preparation for his examinations, this book has also developed in a large measure as an *aide de memoire* and book of reference for practical engineers, besides giving many constructions that may be new to some.

The plates contain illustrations of cylindrical, water-tube, locomotive, donkey and launch boilers, with all their details. There are drawings of furnaces, bracing, stacks, of all usual fittings, as gauges, reducing valves, hydrokineters, circulators; various methods of burning fuel oil and coal are illustrated. Ash injectors, blowing engines, the Howden and the Ellis & Eaves system of forced draft are shown and feed-water filters and heaters are not forgotten.

The construction of the engines is fully illustrated by full details of all parts, and also by several examples of engines in actual use, and from the designs of notable builders. They embrace all types, and vary from a small launch engine to the

engines of the *Deutschland*. Our much-admired framing of the engines of our destroyers is given a prominent place. There are full diagrams of crank efforts, inertia, indicator diagrams, theoretical, actual and combined.

There are full drawings of the necessary auxiliaries, many types of pumps, ejectors, turning and steering engines, ash ejectors, etc.; in fact, all that is necessary for an engineer of a vessel to be familiar with.

Of no little interest are the drawings of various gas and oil engines for launches, as well as the principal systems of steam turbines.

The entire work bristles with interesting data and is a most valuable adjunct to any engineer's library.

The authors deserve great credit for the selection of the most valuable material on the subject of marine engineering.

SCHIFFSKESSEL. By WALTER MENTZ, professor in the Royal Technical School of Danzig. Munich and Berlin: R. OLDENBOURG, 1907.

This book of 300 pages, treating of the development, the design, the construction and the arrangement of boilers on board ship, is a valuable addition to the literature on the subject of marine boilers. The portion devoted to fire-tube boilers is very comprehensive, and the illustrations excellent. That treating of water-tube boilers gives the Schulz-Thornycroft in considerable detail, but only short descriptions, with illustrations, of other Continental and British boilers, while mention is made of only two boilers of American design, the Babcock & Wilcox and the Stirling, the latter of which, as stated by the author, is not in use afloat. Notwithstanding this omission, and the disappointing nature of the chapters devoted to water-tube boilers, the book possesses decided merit. The text is in German, and the measurements in metric units.

OBITUARY.

It is with much regret that the Council announces the death of one of its members since the last issue of the JOURNAL.

Rear Admiral WILLIAM A. WINDSOR, U. S. N. (Retired), died on August 30, 1907.

Admiral Windsor was born in Virginia and was appointed a Third Assistant Engineer in the Navy in 1862. He was transferred to the retired list September 16, 1902, on his own application, after forty years' service.

NOTES OF THE SOCIETY.

A regular meeting of the Society was held at the Navy Department, October 1st, at which nominations were made for officers of the Society for the coming year, as follows:

For President.

Commander W. M. PARKS, U. S. N.
Commander R. S. GRIFFIN, U. S. N.

For Secretary-Treasurer.

Commander THEO. C. FENTON, U. S. N., Retired.
Lieutenant J. B. GILMER, U. S. N.

For Members of Council.

(Three to be selected.)

Commander F. C. BIEG, U. S. N.
Commander B. C. BRYAN, U. S. N.
Commander H. P. NORTON, U. S. N.
Engineer-in-Chief CHAS. A. MCALLISTER, U. S. R. C. S.
Commander W. W. WHITE, U. S. N., Retired.
Lieutenant Commander C. W. DYSON, U. S. N.
Lieutenant J. B. GILMER, U. S. N.

Voting slips have been forwarded to all the members, to be returned at once. The votes will be counted at a meeting of the Society to be held at the Bureau of Steam Engineering, Navy Department, at 4.30 P. M., Thursday, December 26, 1907.

AMERICAN SOCIETY OF NAVAL ENGINEERS—1907.

OFFICERS FOR 1907.

President:

Commander B. C. Bryan, U. S. Navy.

Secretary and Treasurer:

Commander Theo. C. Fenton, U.S. Navy, Retired.

Council:

Captain A. F. Dixon, U. S. N.
Commander R. S. Griffin, U. S. N.
Commander B. C. Bryan, U. S. N.
Gommander H. P. Norton, U. S. N.
Commander Theo. C. Fenton, U. S. N., Retired.

HONORARY MEMBERS.

The Secretary of the Navy.................................Navy Department, Washington, D. C.
The Assistant Secretary of the Navy..........................Navy Department, Washington, D. C.
Charles Whiteside Rae, Rear Admiral, U. S. N., Engineer-in-Chief,
Navy Department, Washington, D. C.
Chas. H. Haswell, Consulting and Superintending Engineer,
324 West 78th street, New York, N. Y.
B. F. Isherwood, Chief Engineer, U. S. N. (retired)........111 East 36th street, New York, N. Y.
Geo. W. Melville, Rear Admiral, U. S. N. (retired)............615 Walnut street, Philadelphia, Pa.
George R. Dunell, Esq ....33 Spencer Road, Chiswick W., England.

Prize Essayists.

W. W. White (1897), Commander, U. S. N. (retired),
Bureau Steam Engineering, Navy Department, Washington, D. C.
W. F. Durand (1898) ....Stanford University, California.
B. C. Ball (1899) ....Willamette I. & S. Works, Portland, Oregon.
W. F. Worthington (1900), Commander, U. S. N............U. S. Naval Academy, Annapolis, Md.
J. R. Edwards (1901), Commander, U. S. N..................................Navy Yard, Portsmouth, N. H.
F. W. Bartlett (1902), Commander, U. S. N.........Midvale Steel Co., Nicetown, Philadelphia, Pa.
Charles W. Dyson (1903), Lieutenant Commander, U. S. N.,
Bureau Steam Engineering, Navy Department, Washington, D. C.
Janson, Ernest N. (1904), Mechanical Engineer.............Navy Department, Washington, D. C.

MEMBERS.

Addicks, W. R., Vice President Consolidated Gas Company...No. 4 Irving Place, New York, N. Y.
Allderdice, W. H., Commander, U. S. N. (retired).
Allen, F. B., Vice-Pres. Hartford Steam Boiler Insp. and Ins. Co.,
Residence, 61 Willard street, Hartford, Conn.
Anderson, M. A., Commander, U. S. N. (retired)..................834 Prospect Place, Madison, Wis.
Arnold, Solon, Commander, U. S. N. (retired).
Ayres, S. L. P., Chief Engineer, U. S. N. (retired)................1420 Master st., Philadelphia, Pa.

Bailey, F. H., Commander, U. S. N.........Navy Yard, New York, N. Y.
Baird, G. W., Rear Admiral, U. S. N.......1505 Rhode Island avenue, N. W., Washington, D. C.
Baker, Frederick W.........Care Lake Torpedo Boat Co., Bridgeport, Conn.
Baldwin, F. P., Lieutenant, U. S. N.
Ball, Walter, Lieutenant Commander, U. S. N.
Barrows, H. C., Chief Engineer, U. S. R. C. S.........U. S. S., *Boutwell*, Newbern, N. C.
Barry, Ralph E.........182 Washington Park, Brooklyn, N. Y.
Barton, J. K., Commander, U. S. N.........U. S. Naval Academy, Annapolis, Md.
Bass, I. E., Lieutenant, U. S. N.
Bates, A. B., Rear Admiral, U. S. N. (retired).........18 Riverside Drive, Binghamton, N. Y.
Bayley, W. B., Captain, U. S. N.
Beach, E. L., Lieutenant Commander, U. S. N.
Bennett, F. M., Commander, U. S. N.
Bennett, K. M., Lieutenant, U. S. N.
Bieg, F. C., Commander, U. S. N.
Bowers, F. C., Lieutenant Commander, U. S. N.
Bowers, J. T., Lieutenant, U. S. N.
Boyd, David F., Lieutenant, U. S. N.
Brady, John R., Lieutenant, U. S. N.
Bray, Chas. D., Professor Civil and Mechanical Engineering.........Tufts College, Mass.
Brinser, H. L., Lieutenant, U. S. N.
Brooks, LeRoy, Jr., Ensign, U. S. N.
Brooks, W. B., Chief Engineer, U. S. N. (retired).........437 West 6th street, Erie, Pa.
Bryan, B. C., Commander, U. S. N.,
Bureau Steam Engineering, Navy Department, Washington, D. C.
Bryan, J. I., 1st Assistant Engineer, U. S. R. C. S.........U. S. S. *Penrose*, Pensacola, Fla.
Bulmer, Roscoe C., Lieutenant, U. S. N.
Burke, W. S., Passed Assistant Engineer, U. S. N. (retired),
60 Buckingham street, Cambridge, Mass.
Bush, W. W., Lieutenant Commander, U. S. N.
Butler, H. V., Lieutenant, U. S. N.

Cage, H. K., Lieutenant, U. S. N.
Capps, W. L., Rear Admiral, Chief Constructor, U. S. N., Navy Department, Washington, D.C.
Carr, C. A., Commander, U. S. N.
Carney, R. E. Assistant Engineer, U. S. N. (retired).
Carter, T. F., Lieutenant Commander, U. S. N.........Navy Yard, Pensacola, Fla.
Castle, Guy W. S., Ensign, U. S. N.
Cathcart, W. L.........Hoyt, Montgomery County, Pa.
Chambers, W. H., Commander, U. S. N. (retired),
Chandler, Lloyd H., Lieutenant Commander, U. S. N.,
U. S. S. *Connecticut*, care Postmaster, New York.
Chappell, R. H.........Bureau Engraving and Printing, Washington, D. C.
Clark, T. W.........Care Standard Oil Co. of New York, 4 Des Voeux Road, Hong Kong, China.
Cluverius, W. T., Lieutenant, U. S. N.
Cole, W. C., Lieutenant Commander, U. S. N.
Collins, Henry L.........850 Ridge avenue, Allegheny, Pa.
Cone, Hutch I., Lieutenant, U. S. N.
Connelly, Louis J., Lieutenant, U. S. N.
Cook, Allen M., Lieutenant, U. S. N.
Cooley, Mortimer E., Professor Mechanical Engineering,
University of Michigan, Ann Arbor, Mich.
Cooper, I. T., Lieutenant, U. S. N. (retired).........Camden, Del.
Cothran, W. B., Warrant Machinist, U. S. N.
Cowles, W. Barnum, Constructing Engineer.........Cor. Lake and Wason streets, Cleveland, Ohio.
Cox, Leonard M., Chief Engineer, L. H. & St. L. Ry. Co.........Louisville, Ky.
Coxe, W. G.........Harlan-Hollingsworth Corporation, Wilmington, Del.
Crank, R. K., Lieutenant Commander, U. S. N.
Crawford, Robt., Passed Assistant Engineer, U. S. N. (retired),
Wm. Cramp and Sons, Beach and Ball streets, Philadelphia, Pa.

Creighton, W. H. P., Assistant Engineer, U. S. N. (retired),
1438 Henry Clay Avenue, New Orleans, La.
Crenshaw, A., Lieutenant, U. S. N.
Cronan, W. P., Lieutenant, U. S. N.
Cunningham, A. C., Civil Engineer, U. S. N.
Bureau of Yards and Docks, Navy Department, Washington, D. C.
Curtis, C. J., 2d Assistant Engineer, U. S. R. C. S. ....U. S. S. *Apache*, Baltimore, Md.

Danforth, Geo. W., Lieutenant, U. S. N. (retired)..........Union Iron Works, San Francisco, Cal.
Davis, E. W., Engineer, U. S. R. C. S.........................U. S. S. *Arcata*, Post Townsend, Wash.
Davis, Norris K.................................S. W. corner 7th and Harrison sts., San Francisco, Cal.
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VOL. XIX. NO. 4.

JOURNAL

OF THE

American Society of Naval Engineers.

NOVEMBER, 1907.

PUBLISHED QUARTERLY BY THE SOCIETY.

WASHINGTON, D. C.
R. BERESFORD, PRINTER, 618 F STREET, N. W.
1907.

Entered at the Post office at Washington, D. C., as Second Class Matter.

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The subscription price of the Journal is $5.00 per annum, payable in advance.

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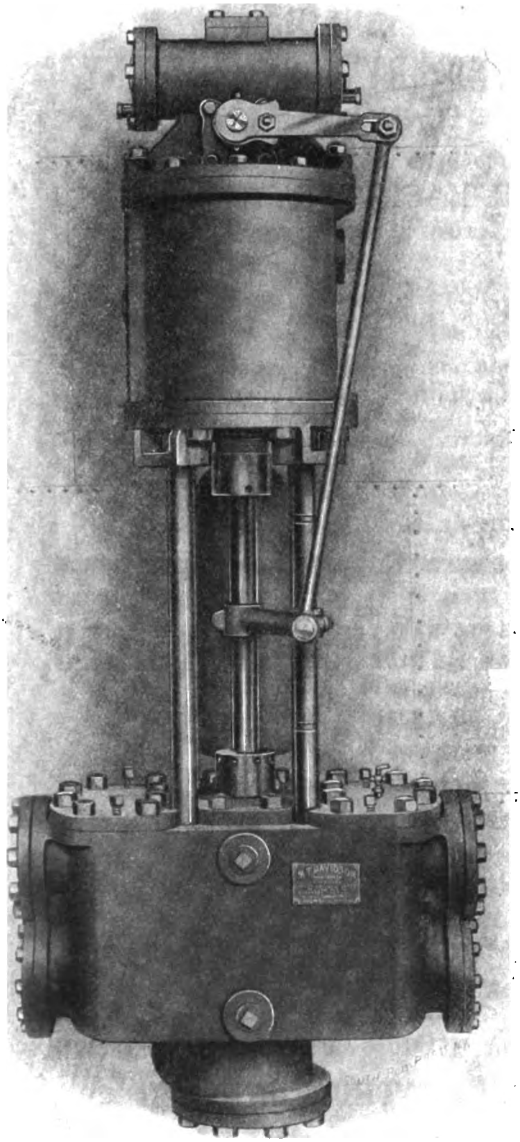

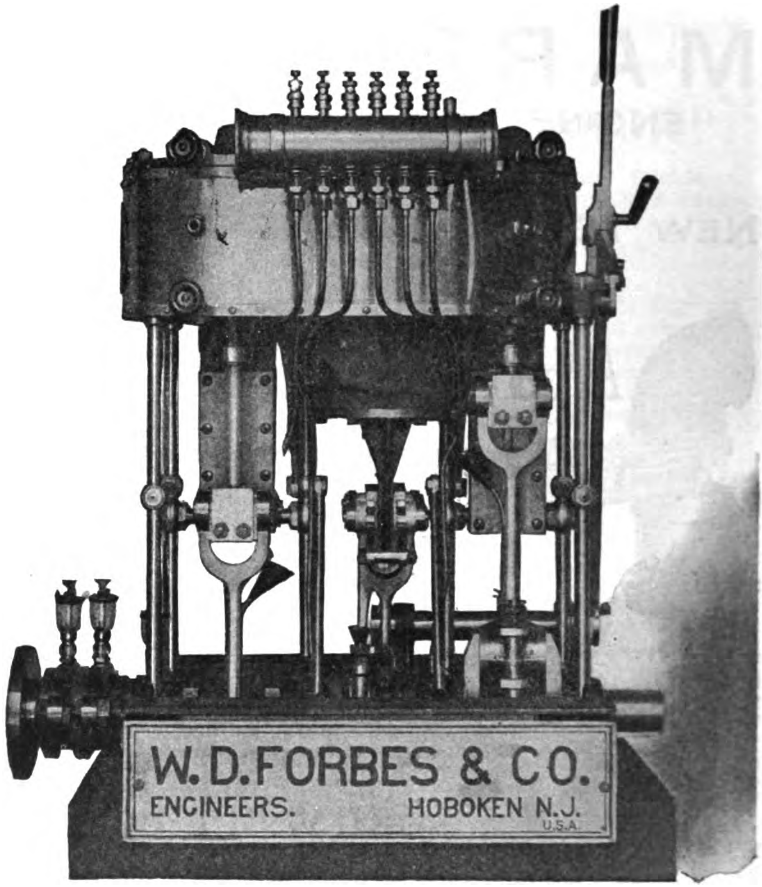

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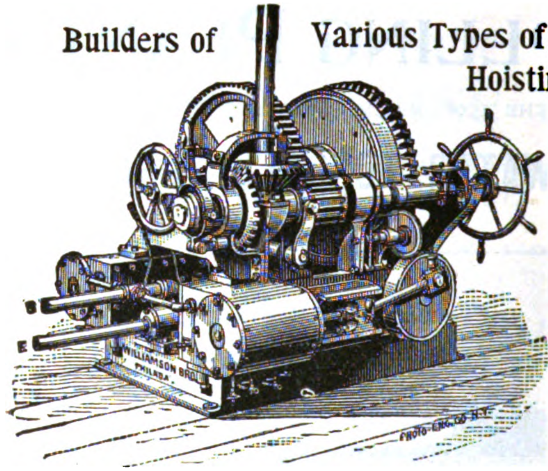

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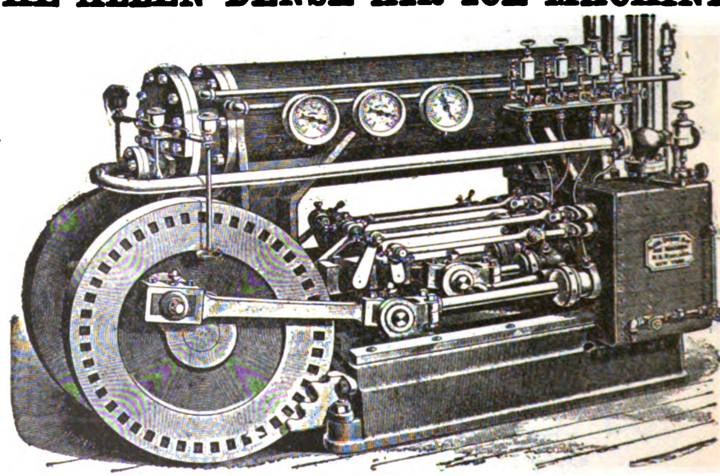

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